Gas Processing

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MODULE 1:

Basic Process Principles

BASIC PROCESS PRINCIPLES

In the pages that follow, we will discuss many different sweetening processes. These processes involve different materials for removing sour gas constituents from the gas stream, different techniques of regeneration and different principles of operation. In spite of their many and varied differences, however, there are certain underlying fundamentals of physical and chemical operations that are similar and, in many cases, the same. Before beginning a study of the different sweetening processes, we should probably review the basic principles and underlying fundamentals of the many processes. The review of these fundamentals presented below is necessarily brief. For a more complete and detailed treatment of these fundamentals the reader is referred to "Gas Conditioning and Processing" by Campbell. (2.1)

Basic Dimensions

In all processes, we are required to express quantities in terms of certain fundamental dimensions. These fundamental dimensions are more or less arbitrary assignments of value from which many derived dimensions and quantities can be developed. In most cases, the fundamental dimensions are considered to be length (L), time (0), mass (m), force (f) and temperature (r). Table 2.1 summarizes the conversion between these fundamental dimensions for several sets of common units.

MASS AND ENERGY BALANCES

Two of the most powerful tools the engineer has in his bag of tricks are the material or mass balance and the energy or heat balance. The material balance can be used to determine the production from a plant, the necessary flow rate of treating
Solution, the size of pump required and many other variables. The energy balance can be used to determine the heat input required in a reboiler, the horsepower of a pump or the heat duty of a heat exchanger.

Common Systems of Units

Systems Units of	Engineering English	Absolute English	Technical English	Absolute metric		Engineering meric
				Cgs	Mks	
Mass	Pound mass ($1b_m$)	Pound mass ($1b_m$)	Slug	Gram (gm_m)	Meter (m)	Centimeter (cm)
Length	Foot (ft)	Foot (ft)	Foot (ft)	Centimeter (cm)	Meter (m)	Centimeter (cm)
Time	Second (sec)	Second (sec)	Second (sec)	Second (sec)	Second (sec)	Second (sec)
Force	Pound Force ($1b_f$)	Poundal (pdl)	Pound Force ($1b_f$)	Dyne	Newten (nt)	Lilogram Force ($kilo_f$)
Temperature	Degree Rankine (R^o)	Degree Rankine (R^o)	Degree Rankine (R^o)	Degree Kelvin (k^o)	Degree Kelvin (k^o)	Degree Kelvin (k^o)
Value of R	1545.33 $\frac{(ft)(1b_f)}{\left[\frac{1b-}{mole}\right](R^O)}$	$4.9719x10^4$ $\frac{(ft)(pdl)}{\left[\frac{1b-}{mole}\right](R^o)}$	48.031 $\frac{(ft)(1b_f)}{\left[\frac{slug-}{mole}\right](R^O)}$	$8.3144x10^7$ $\frac{(dyne)(cm)}{\left[\frac{gm-}{mole}\right](k^o)}$	$8.3144x10^7$ $\frac{(nt)(m)}{\left[\frac{kilo-}{mole}\right](k^o)}$	$8.4783x10^4$ $\frac{(kilo)(cm)}{\left[\frac{kilo-}{mole}\right](k^o)}$
Value of R	32.1739 $\frac{(1b_m)(ft)}{(1b_f)(sec^2)}$	1.0 $\frac{(1b_m)(ft)}{(1b_f)(sec^2)}$	1.0 $\frac{(slug)(ft)}{(1b_f)(sec^2)}$	1.0 $\frac{(gm_m)(cm}{(dyne)(sec^2}$	1.0 $\frac{(kilo_m)(m)}{(nt)(sec^2)}$	1.0 $\frac{(kilo_m)(m)}{(nt)(sec^2)}$

Table 2.1 Summary of Common Systems of Units (Ref. 2.1)

In order to properly use and apply heat and material balances, we must understand the thermodynamic definitions of the system and the surroundings. The system is very simply defined as that portion pf the universe in which we are interested and concerned at a given time. The surroundings are all other parts of the universe save the system, though ordinarily we are most concerned with that part of the surroundings adjacent to the system. Mass and energy are considered to be "in" or "out" of the system any time one or the other crosses the arbitrarily defined boundary of the system.

Mass Balances

For a given defined system, we can express the mass balance for a given period as:
Mass change in system in given time period = (mass in during that time period) - (mass out during that time period) ± (mass gained or lost by atomic transformation)
In symbolic form the mass balance can be expressed as

$$M_E - M_B = \sum M_I - \sum M_O \pm M_P \qquad (2.1)$$

Where:

$M_E - M_B$ = mass change in system in time period
$\sum M_I$ = summation of all mass entering the system in the time period
$\sum M_O$ = summation of all mass leaving the system in the time period
M_P = gain or loss in mass from atomic transformation

In the processes that we will consider M_P, the gain or loss in mass caused by atomic transformation, will be zero.
A mass balance can be written based on the total amount of material crossing the boundaries of the system. In this case, it is commonly referred to as an overall material balance. In some instances there is advantage to writing a material balance for one of the components in the streams entering or leaving the system. In this case the balance is referred to as a component material balance. There are even instances when there is good reason for writing a material balance based on one atomic species such as the sulfur in ILS and other compounds entering or leaving the sweetening process.

Energy Balances

We can express the form of the energy balance as:

Energy change in system in a given time period = (energy into the system during that time period) - (energy out of the system during that time period) ± (energy gained or lost because of atomic transformation).

Before we can write the energy balance in equation form, we must define the forms of energy to be considered in the energy balance. The total energy of the system is arbitrarily defined as the sum of three quantities: the potential energy (PE); the kinetic energy (KE); and the internal energy (U). The potential energy of the system is defined as the potential the system has for doing work because of its elevation above a given datum plane. The kinetic energy of the System is defined as the capacity the system has for doing work because of motion relative to a fixed point. The internal energy is arbitrarily defined as all energy possessed by the system except that defined as potential and kinetic energy.

There are two other energy terms that need to be defined before completely accounting for the energy balance around a system. These are heat and work.

Heat (Q) is defined as all energy crossing the boundary of the system solely as a result of a temperature difference between the system and surroundings that is in no way associated with transfer of mass.

Heat gained by the system is arbitrarily defined as positive (+) and that lost by the system negative (-).

Work (W) is defined as all non-mass associated energy crossing the boundary of the system by any mechanism other than a temperature differential. Work done by the system is defined as positive (+) and work done on the system is negative (-).

The total energy associated with mass entering or leaving the system is equal to (H + PE + KE). Enthalpy (H) is a conceptual term which is defined as H = U + PV.

Using these definitions of terms we can write the energy balance for the system in symbolic form as:

$$(U + PE + KE)_E - (U + PE + KE)_B = \sum (H + PE + KE)_I - \sum (H + PE + KE)_O + \sum Q - \sum W - E_C \qquad (2.2)$$

As before, the energy gained or lost by atomic transformation (E) is negligible for the cases we will be considering.

FLUID FLOW

In calculations concerning the flow of liquids through pipe and tubing we are concerned with two interrelated problems - the pressure drop of the fluid in flowing through the line and/or the horsepower of the pump required to pump the liquid through the line. For isothermal and adiabatic flow of any fluid through a line containing no work or heat source an energy balance can be written as follows:

$$\sum v dp + \Delta x + \frac{\Delta(U^2)}{2g_c} = -W_f - W \qquad (2.3)$$

Where:

v = fluid specific volume in cubic feet per pound
P = absolute pressure in pounds per square foot *
x = elevation above datum point in feet
u = fluid velocity in feet per second
W_f = friction loss in feet per second
W = work done by the fluid g = gravitational constant in

All of the terms involved in Equ. 2.3 must be expressed in common units. The most convenient and widely accepted method for expressing each of the terms is in the equivalent height of liquid for each term. Actually the dimensions of each term become $\frac{lb_f - ft}{lb_m}$.Considering lb_m and lb_f to be equal numerically and equivalent dimensional, leads to the colloquial expression "feet of fluid" as a means of expressing the different terms.

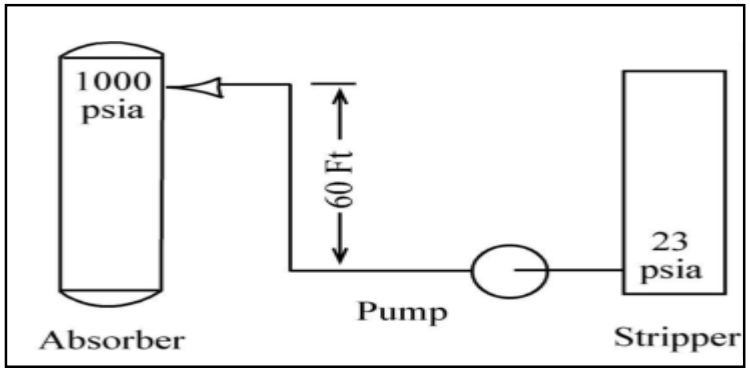
For incompressible fluids (liquids) where there is little or negligible change in density with pressure, Equ. 2.3 can be written as

$$\frac{\Delta p}{p} + \Delta X + \frac{\Delta(u^2)}{2g_C} = -W_f - W \qquad (2.3a)$$

Where: ρ = liquid density, Ib/cu ft

Proper application of Equ. 2.3 Or 2.3a will be clarified by study of the following example.

Example



As shown in the sketch above, a 20% monoethanolamine (MEA) solution is being pumped from the bottom of the stripper to the top of the absorber. The bottom pressure on the stripper is 23 psia and the top pressure on the absorber is 1000 psia. The top of the absorber is 60 feet above the level of the pump. Write the overall energy balance for this system.

Solution

We can evaluate the individual terms in the energy balance equation as follows:

$$\frac{\Delta P}{\rho} = \frac{100 - 23}{\rho} \text{ X } 144$$

The density of the MEA solution can be determined from Figure 2.1. If the flowing temperature of the amine solution is assumed to be 100° F., we find that the density is 0.999 gm/cc. In lb/cu ft this will be 0.999 x 62.4 = 62.34 lb/cu ft. $\frac{\Delta P}{P}$ then becomes

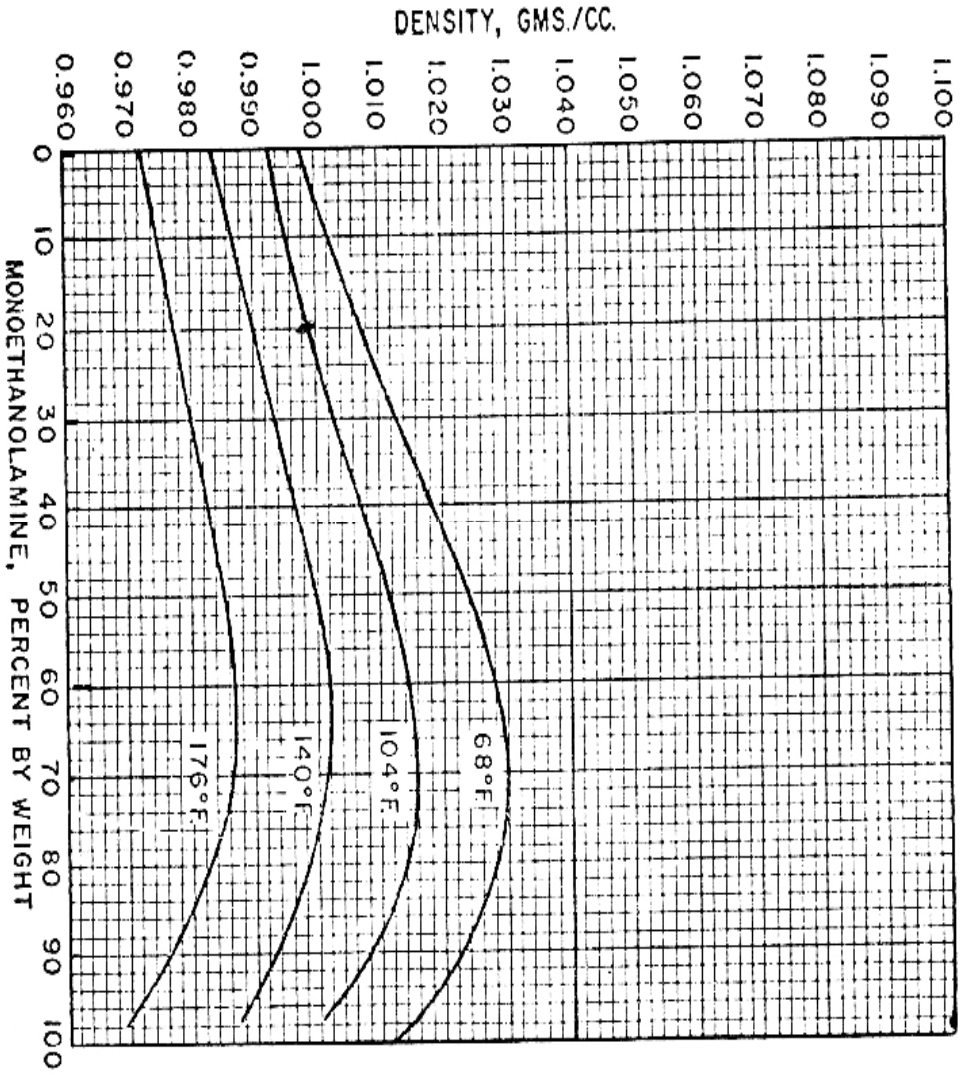


Fig. 2.1

Density of MEA Solutions (Ref. 2.5. Courtesy Dow Chemical Co.)

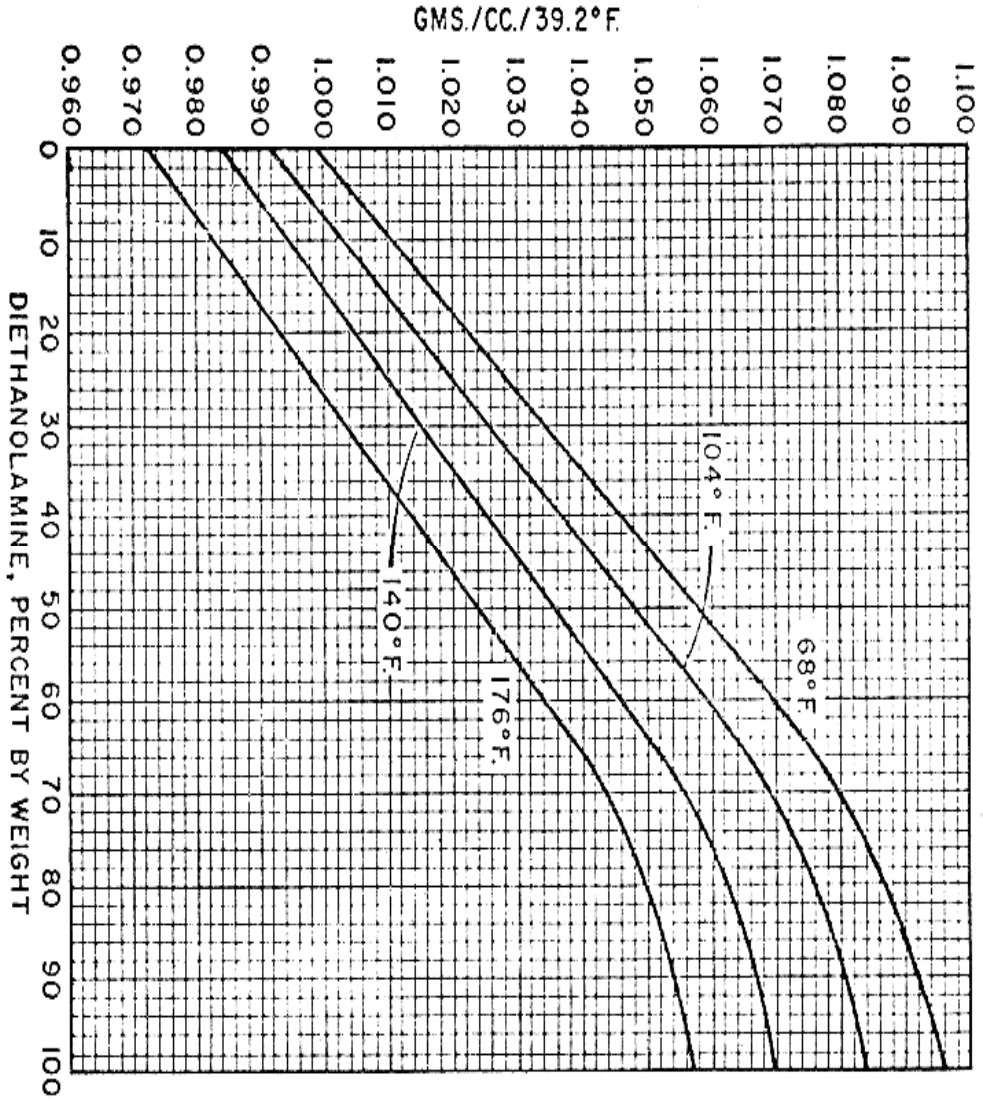


Fig. 2.2
Density of DEA solutions (Ref. 2.5. Courtesy Dow Chemical Co.)

$$\frac{\Delta P}{\rho} = \frac{977 \times 144}{62.34} = 2257 \ \frac{1b_f - ft}{1b_m}$$

The difference in elevation will be the difference in level between the pump discharge and the top of the absorber, which was specified to be 60 feet.

In most cases, the velocity terms are so small as to be negligible. Unless there is a change in line size or the line discharges into a tank, the velocity term is 0.

The two remaining terms in Equ. 2.3a, W_f and W cannot be evaluated at this point. There is insufficient information. The final equation as written then would be:

$$2257 + 60 + 0 = -W_f - W$$

The implications of this are that, according to the definitions given earlier for work, the pump must do work on the amine solution in order to pump it to the higher pressure. This, of course, is perfectly logical and reasonable and does explain the negative sign which is certain to appear on the external work term in Equ. 2.3a as written for this problem.

Friction Loss

The friction loss term in Equ. 2.3 and 2.3a includes essentially all of the irreversibility effects for the fluid flowing in the line. It can be expressed as:

$$W_f = \frac{2fU^2L}{g_C D} \qquad (2.4)$$

Where:

f = friction factor, dimensionless
u = fluid velocity in ft/sec.
L = equivalent length of straight pipe in feet
D = inside diameter of pipe in feet

The "friction factor" f has been proven to be a function of the Reynolds number $\frac{(DU\rho)}{\mu}$ the roughness of the pipe wall and the type of flow (laminar, turbulent or transition). In nearly all applications of industrial interest, the flow will be in the turbulent regime (Reynolds number greater than 4,000). Variation of friction factor, f, with Reynolds number is shown in Figure 2.3. The Reynolds number is a dimensionless number, which means that all terms used in calculating the Reynolds number must be in consistent units.

Determining the friction loss using Fig. 2.3 requires calculating the Reynolds number, evaluating the friction factor, and calculating the friction loss from Equ. 2.4. In terms of readily available numbers, the Reynolds number can be calculated as:

$$\text{Re} = \frac{3150(g)(s.c)}{(d)(\mu)} \qquad (2.5)$$

Where:

g = liquid flow rate in gallons per minute
S.G.= liquid specific gravity referred to water
d = inside diameter of pipe, inches
μ = liquid viscosity in centipoises

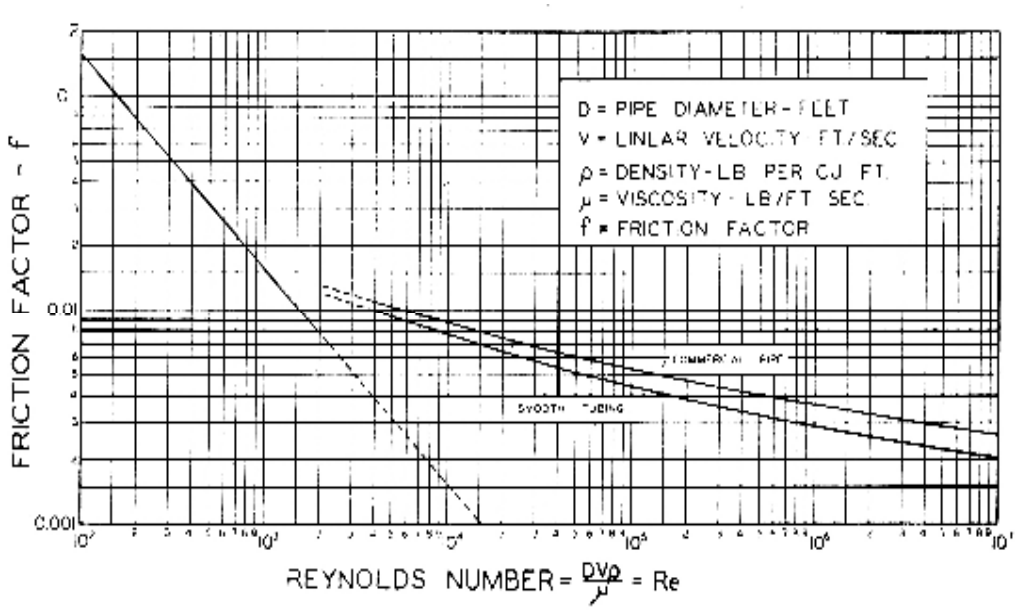

FRICTION FACTOR - f
D = PIPE DIAMETER - FEET
V = LINEAR VELOCITY FT/SEC
ρ = DENSITY - LB PER CU FT
μ = VISCOSITY - LB/FT SEC
f = FRICTION FACTOR
SMOOTH TUBING
0.001
0.01
REYNOLDS NUMBER = DVρ/μ = Re

Fig. 2.3

Friction Factor as. A Function of Reynolds Number

Other versions of Fig. 2.3 are available. Some of these use different forms of Equ. 2.4. This means that one must be careful to use the graph for determining friction factor that corresponds to the form of Equ. 2.4 used for calculating friction loss. In some instances the effect of wall roughness is included in terms of a relative roughness factor. This factor usually relates the "height" of the average roughness to pipe diameter. As a practical matter, the two curves showing smooth tubing and commercial pipe on Fig. 2.3 bracket the range of pipe conditions usually found.

Example Problem

150 gpm of a 25% diethanolamine solution is being pumped through a 3 inch inside diameter pipe. If the flowing temperature of the amine is 120° F., calculate the pressure drop in the line per 100 feet of pipe.

Solution

For a 25% DBA solution:

S.G. = 1.012 (From Fig. 2.2)

μ = 1.3 cp (From Fig. 2.5)

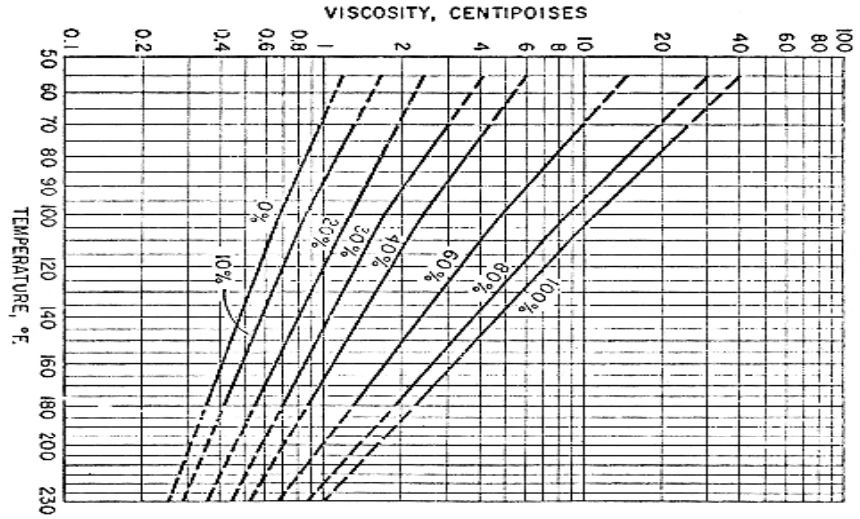


Fig. 2.4
Viscosity of aqueous MEA solutions as function of weight percent MEA
(Ref. 2.5, Courtesy Dow Chemical Co.)

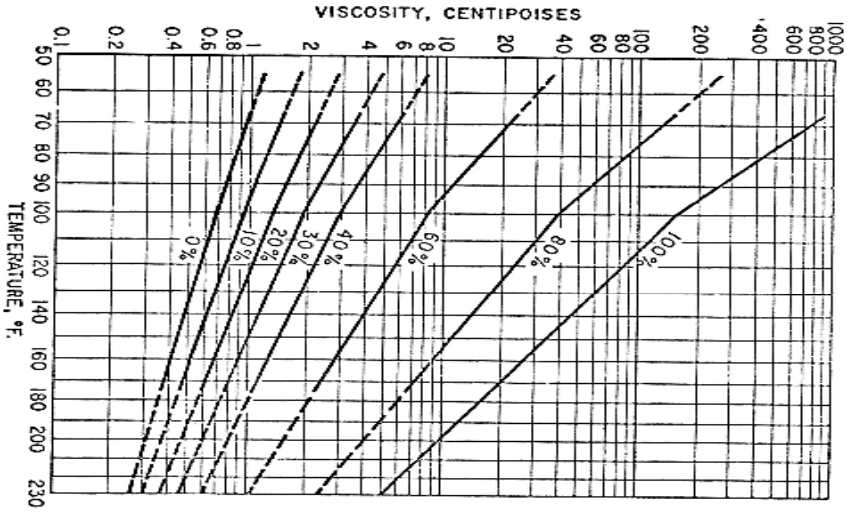


Fig. 2.5
Viscosity of aqueous DEA solutions as function of weight percent DEA
(Ref. 2.5, Courtesy Dow Chemical Co.)

The Reynolds number then is

$$\text{Re} = \frac{3150(150)(1.012)}{(3.0)(1.3)} = 122.600$$

From Fig. 2.3 using the line for commercial pipe, we read a friction factor of .0051. We must now determine the velocity in the line.

$$\text{u} = \frac{150}{7.48} \times \frac{1}{60} \times \frac{4 \times 144}{\pi(3.0)^2} = 6.8 \text{ ft/sec}$$

We can now determine the friction loss as

$$\text{W}_\text{f} = \frac{2(0.0051)(6.8)^2(100)}{32.2(3/12)} = 5.86 \ \frac{ft - lb_f}{lb_m}$$

The pressure drop per 100 feet of line is calculated from the head loss and the fluid density as follows:

$$\Delta \text{p} = \frac{(\Delta h)(\rho)}{144} = \frac{5.86 \times (1.012 \times 62.4)}{144} = 2.57 \text{ psi/100 ft}$$

Equ. 2.4 and Fig. 2.3 can be solved nomographically. When doing this, some simplifying assumption generally is made. Fig. 2.6 is a modification of the nomograph presented by Lewis. (2. 2) His assumption is that the friction factor is constant at a value of 0.004. Fig. 2.6 can be used as described below for several different friction loss cases of interest to the process engineer.

1. To find the pressure drop in a given line for a specified flow rate:
   A. Locate a working point on the reference line by laying a straight edge from the density scale to the flow scale.
   B. Move horizontally from the working point to the diagonal line representing the correct line size.
   C. Read the pressure drop as indicated by the vertical lines at this point.

Example

Use Fig. 2.6 to determine the pressure drop for the 25% DEA solution. The pressure drop for this was previously determined using Fig. 2.3 and Equ. 2.4.

#/hr = gpm x 60 x ρ = 150 x 60 x (8.33 x 1.012) = 75,870

From Figure 2.6: ΔP = 2.2

2. To find the line size giving a desired pressure drop:
   A. Locate a working point as described in 1 above.
   B. Move horizontally from the working point to the vertical line showing the desired pressure drop.
   C. Read the line size on the diagonal scale

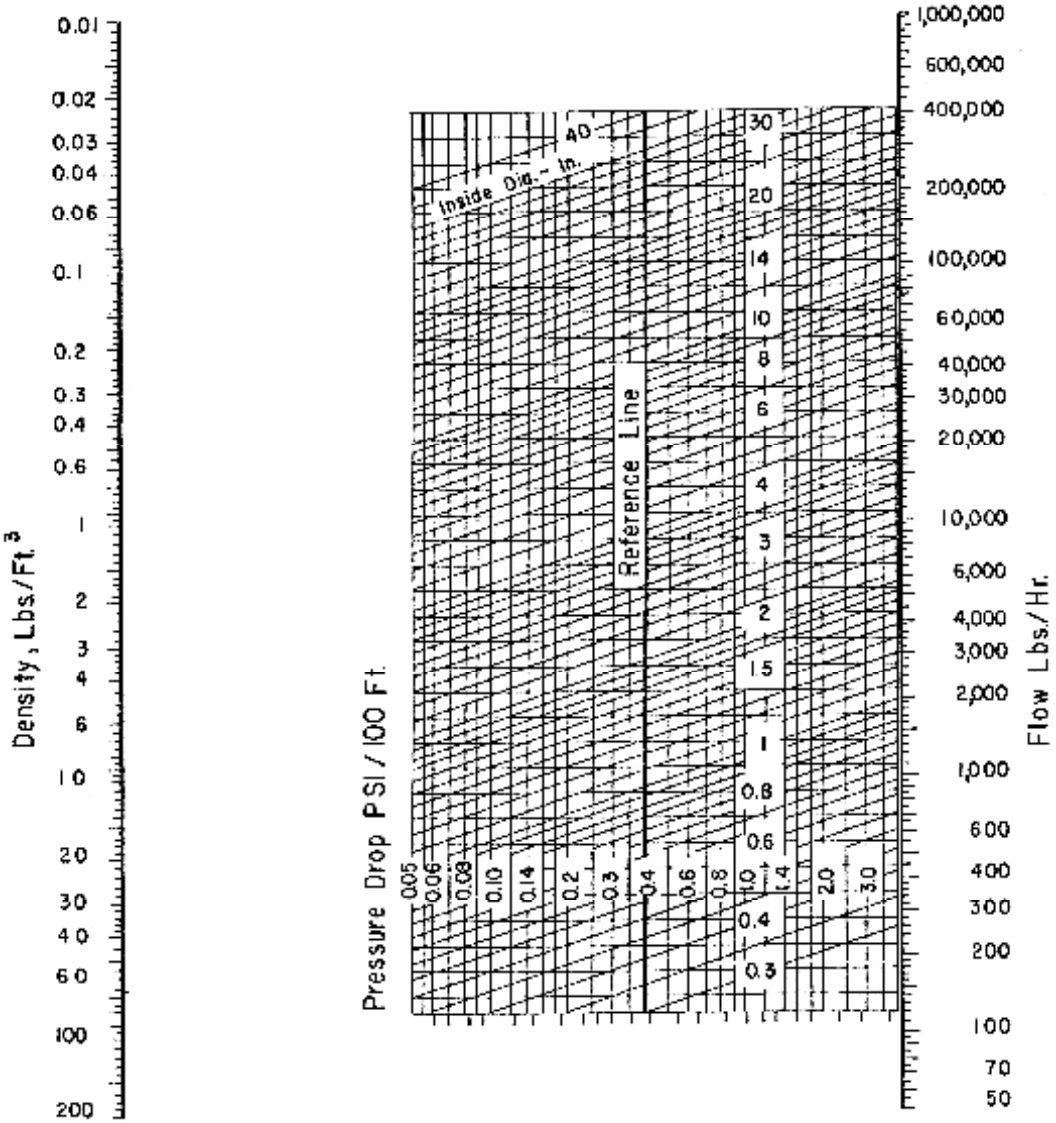


Fig. 2.6

Nomograph for finding line pressure loss (Ref. 2.2. Courtesy Petr. Ref.)

Example problem

Diglycomine is being pumped at a rate of 100 gpm. The pressure drop in the line is 1.2 psi/100 ft of line . if the diglycomine solution is 60% by weight DGA and the flowing temperature 100° F, what is the inside diameter of the line?

Solution

From fig .2.7, S.G. =1.047

#/hr =$100 \times 60 \times (8.33) \times$ 1.0471 = 52.330

ρ = 62.4×1.047 = 65.3

From fig .2.6,i.d.=3.0″

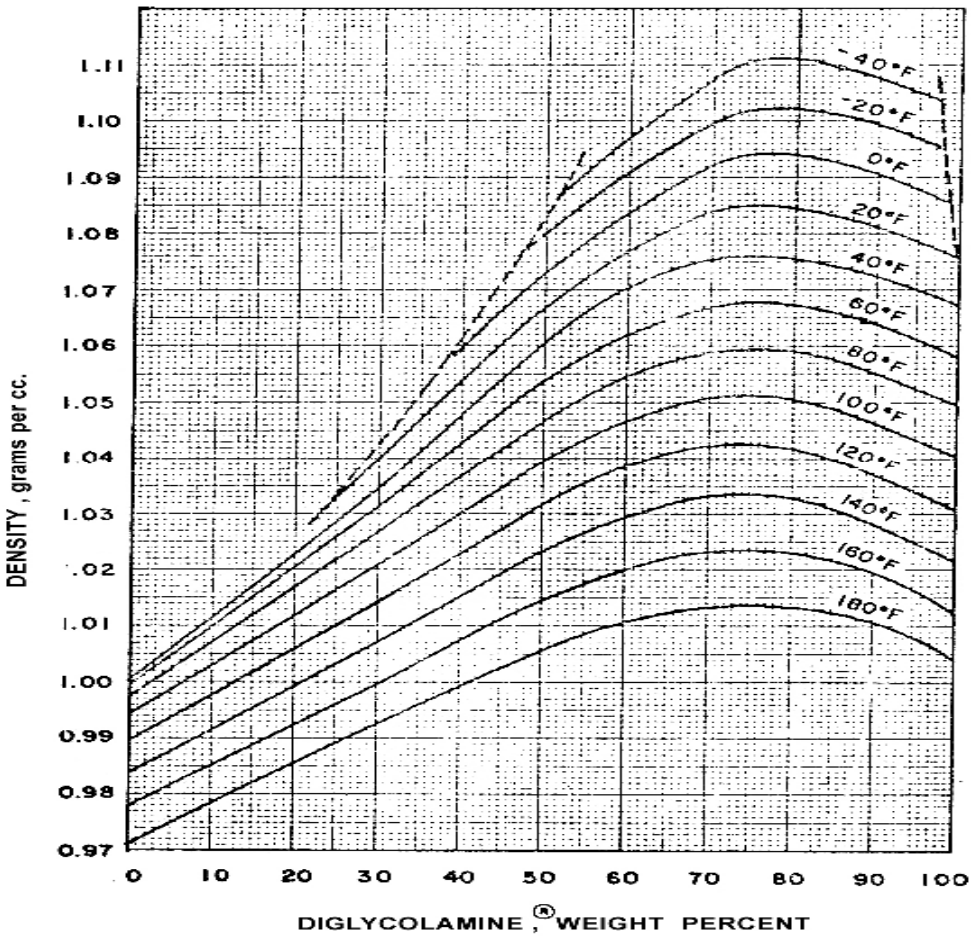


Fig. 2.7
Density of aqueous DGA solutions
(Ref. 2.6. Counrtesy Jefferson Chem. Co.)

3. To correct for a different friction factor:
   A. Find a working point as in case 1 above.
   B. Move diagonally to a value equal to 100 times the new friction factor as shown on the pressure drop scale.
   C. Move horizontally back to the reference line to find the corrected working point.
   D. Continue with steps B and C as in cases 1 and 2 above.

Example Problem
Suppose for the problem in case 1 we wish to adjust the friction factor to its correct value of .0051.

Solution-
Using the working point found in case 1 we would then move along the diagonal pipe diameter line to a value of 0.51 (100 x .0051) and then horizontally to read a pressure drop of 2.7 psi/100 ft of pipe.

4. To find the pressure drop in a new line size:
   A. Find the point on the chart where the old line size intersects the existing pressure drop.
   B. Move horizontally to the new line size and read the new pressure drop

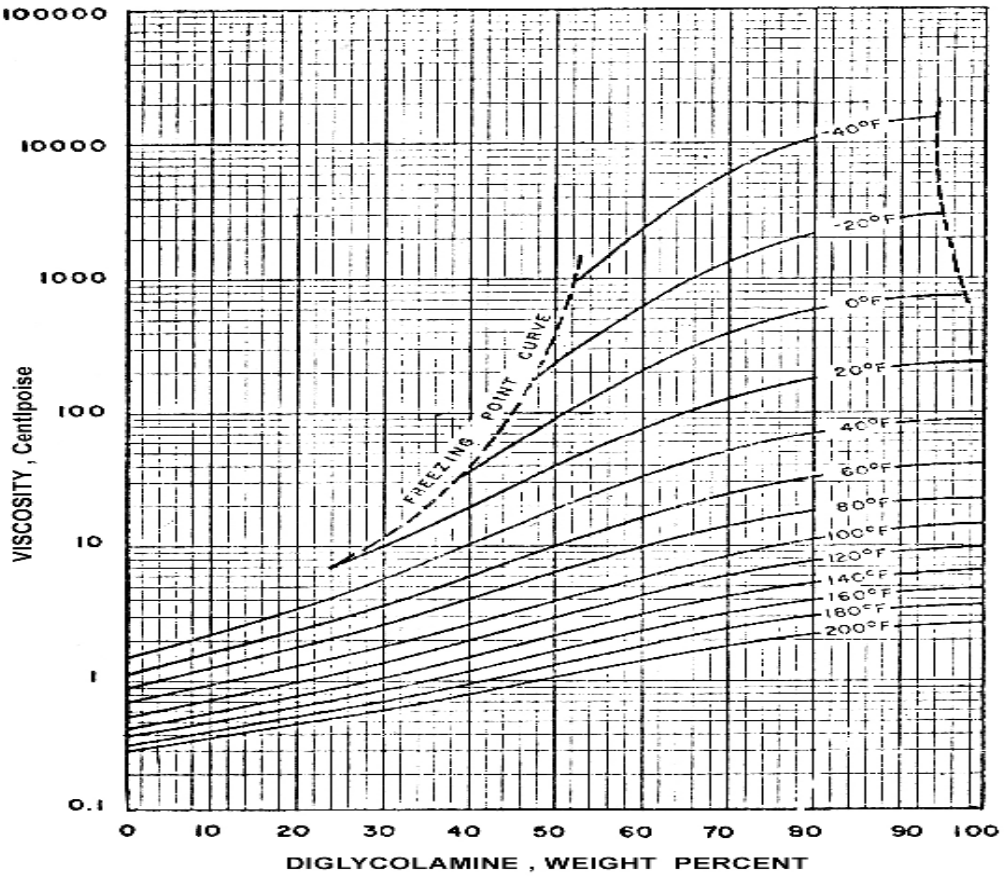


Fig. 2.8 Viscosity of aqueous DGA solutions

(Ref. 2.6.Courtesy Jefferson Chem.Co.)

Example Problem

In the diglycolamine example suppose that we wish to find what the pressure drop would be in a 4 inch diameter line.

Solution

We would move horizontally from the intersection from the 1.2 psi in the 3 inch diameter line to the 4 inch diameter line and read a pressure drop of approximately 0.25 psi per 100 ft of line.

5. To find the allowable flow in a given line:
   A. Use the specified line size and allowable pressure drop to find a point on the chart.
   B. Move horizontally to the reference line to find a working point.
   C. Lay a straight edge from the density scale through the working point and read the allowable flow rate on the flow rate axis.

Example Problem

We wish to find the amount of diglycolamine that can be pumped through a 2 inch inside diameter line with an allowable pressure drop of 2.0 psi/ 100 ft of line.

Solution

Connect the density point (65.3 Ib per cubic ft at 100 F.) with the reference point found from the 2 inch i.d. and 2.0 psi line and read from Fig. 2.6 26,000 Ibs/hr

gpm x 60 x (8.33 x 1.047) = 26,000

gpm = 49.7

Use of Fig. 2.3 and Equ. 2.4, or Fig. 2.6, requires the total equivalent length of "straight" pipe in the line. In order to determine this equivalent length of straight pipe, the valves and fittings in the line must be converted to their equivalent length of straight pipe. This is most commonly done through the use of a nomograph such as Fig. 2.9. As shown, Fig. 2.9 is used by connecting the point on the reference line representing the particular valve or fitting and the appropriate pipe diameter. The equivalent length of the fitting in terms of feet of straight pipe is then read from the middle axis.

Example Problem

Determine the equivalent length of straight pipe for a 2 inch line which contains 150 feet of straight pipe, 2 standard elbows and 1 open globe valve.

Solution

From Fig. 2.9 we can determine the equivalent lengths of the open globe valve and the standard elbows and add these to the length of straight pipe to determine the equivalent length of straight pipe in the line:

Straight pipe		150
2 Standard Elbows	2x5.2	10.4
Open Globe valve		54
Total		214.4 feet

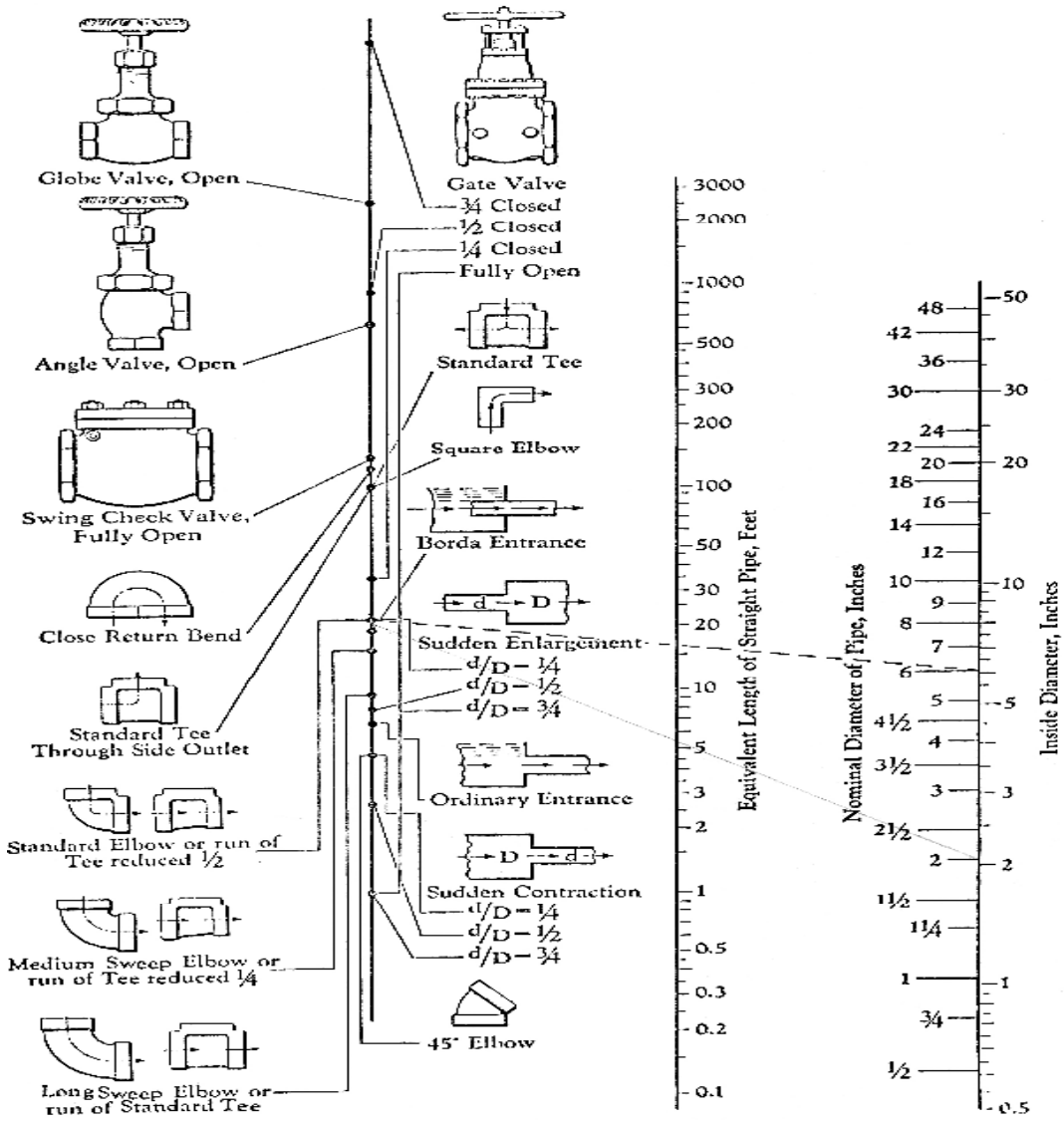


Fig. 2.9

Resistance of valves and fittings to flow of fluids (Ref. 2.9 Courtesy Gulf Publishing Co.)

Horsepower

The horsepower of a pump required for pumping a fluid between two points in a line is generally expressed in terms of the pressure difference across the pump. The required pressure difference is determined from proper application of Equ. 2.3 between the desired points in the pipeline. The pump horsepower required is expressed as:

Where:

QL = the liquid flow in gallons per minute
W = the work requirement in horsepower
$P_2 - P_1$ = pressure gain in pounds per sq ft
E = the pump efficiency

Example Problem

Determine the horsepower required to pump 200 gallons/min of m-pyrole against a total pressure head of 250 psi .

Solution

$$W_h = \frac{200 \times (250 \times 144)}{33.000 \times 7.48}$$

This represents the so-called hydraulic horsepower or the amount of work that must be transmitted to the liquid (in this case the m-pyrole) by the pump. The horsepower of the pump driver must be greater than this because of the efficiency (E) of the pump.

HEAT TRANSFER

Heat transfer is normally expressed in terms of the overall heat transfer coefficient as:

$$Q = uA\Delta t \quad (2.7)$$

where:

Q = rate of heat transfer, Btu/hr
U = overall heat transfer coefficient, BTu/hr, sq ft, F.
A = area through which heat is being transferred in sq ft
Δt = the temperature difference

The temperature difference for heat transfer is usually expressed in terms of the log mean temperature difference at:

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\ln(\frac{\Delta t_1}{\Delta t_2})} \quad (2.8)$$

Example Problem

Lean amine solution enters an amine cooler at 160 F. and is cooled by water to 110° F, Water enters the cooler at 85° F. and is discharged from the exchanger at 105° F. What is the log mean at for the exchanger?

Solution

$$\Delta t = 160 - 205 = 55^o$$

$$\Delta t = 110 - 85 = 25^o$$

$$\Delta t_m = \frac{55 - 25}{\ln 55/25} = \frac{30}{38.1} = 3.18$$

If the area of the exchanger is 600 square feet and the overall heat transfer coefficient is 135 Btu/hr/sg ft °F., how much heat will be transferred from the amine to the water per hour?

$$Q = UA\Delta t = 135 \times 600 \times 38.1 = 3.09 \times 10^6 \text{ Btu/hr}$$

For counter-current flow of two fluids in an exchanger composed of two concentric pipes, Equ. 2.8 applies exactly. For other exchangers such as shell and tube exchangers Δt_m must be corrected for the lack of true counter-current flow in the exchanger The correction is made by multiplying Δt_m by a correction factor F. Charts for estimating F are available in Campbell. (2.1)

Coefficients

In Equ. 2.7 there is an overall heat transfer coefficient U and an area A that are related to each other. In the general case heat will be transferred through three resistances: a liquid film coefficient on one side of the tube wall, the resistance of the tube wall, the resistance of the liquid or fluid film on the other side of the tube wall. The overall film coefficient can be based on any of these areas. The amount of heat transferred through each resistance in series is the same, but the areas of the resistances are different. This means that the overall heat transfer coefficients must be different if the rates of heat transfer per unit of time are to be equal. The overall heat transfer coefficient is related to the individual film coefficients as shown in Equ. 2.9.

$$U_1 = \frac{1}{\frac{1}{h_1} + \frac{A_1 L}{K A_W} + \frac{A_1}{h_2 A_2}} \qquad (2.9)$$

Where:

- h_1 = film coefficient for fluid 1
- h_2 = film coefficient for fluid 2
- K = thermal conductivity of tube wall
- A_1 = surface area of pipe for film h
- A_2 = surface area of pipe for film h
- A_W = average wall area of pipe or tubing
- L = thickness of wall of pipe or tubing

Predicting the individual heat transfer coefficients for use in Equation 2.9 is frequently a difficult and time-consuming task. Also, particularly in the case of shell and tube exchangers, exchanger configuration and geometry have considerable effect on the individual coefficients. Consequently, determination of the individual coefficients is best left to the heat exchange expert who has the physical dimensions of the particular exchanger.

There are ways, however, of estimating typical overall coefficients that can be expected of a heat exchanger in a given service. Table 2.2 tabulates typical overall heat transfer coefficients that can be expected in heat exchangers for a variety of common services. The coefficients shown in Table 2.2 are sufficiently accurate for planning purposes.

In cases where there is no service in Table 2.2 that correlates well with the service of the proposed exchanger, use may be made of Figure 2.10 which relates individual and overall heat transfer coefficients through

$$\frac{1}{U} = \frac{1}{h_p} + \frac{1}{h_t} \qquad (2.10)$$

Where:

U = overall heat transfer coefficient
h_p = coefficient for process fluid P
h_t = coefficient for thermal fluid

Figure 2.10 is obviously an approximation to the overall heat transfer coefficient, but its simplicity and speed of use are compensating factors. In the example shown by the dotted line, an organic liquid is being cooled by cooling tower water. By connecting the appropriate points on the thermal and process axes, the overall coefficient is shown to be approximately 90.

The value selected in the range of individual coefficients should be at the upper limit of the range when: the condensing system is free of enerts; the thermal conductivity of the liquid is known to be high; the viscosity of the liquid is known to be low. The value should approach the lower limit when the viscosity of the liquid is known to be high.

A factor for fouling or scaling of the heat transfer surface is frequently included in the overall heat transfer coefficient.

$$U_1 = \frac{1}{\frac{1}{h_1} + \frac{A_1 L}{K A_W} + \frac{A_1}{h_2 A_2} + F} \qquad (2.11)$$

The factor F in the denominator of Equation 2.11 is to allow for the resistance to heat transfer of the scale that forms in 'many heat transfer services. Values for F are available in many works on heat transfer. Typical values for many services in the gas processing

and sweetening area are presented in Table 2.3. The values given are based on plant data, but not data gathered in a systematic investigation. Consequently, the values in Table 2.3 should be looked upon as guidelines rather than firm values of the fouling factor.

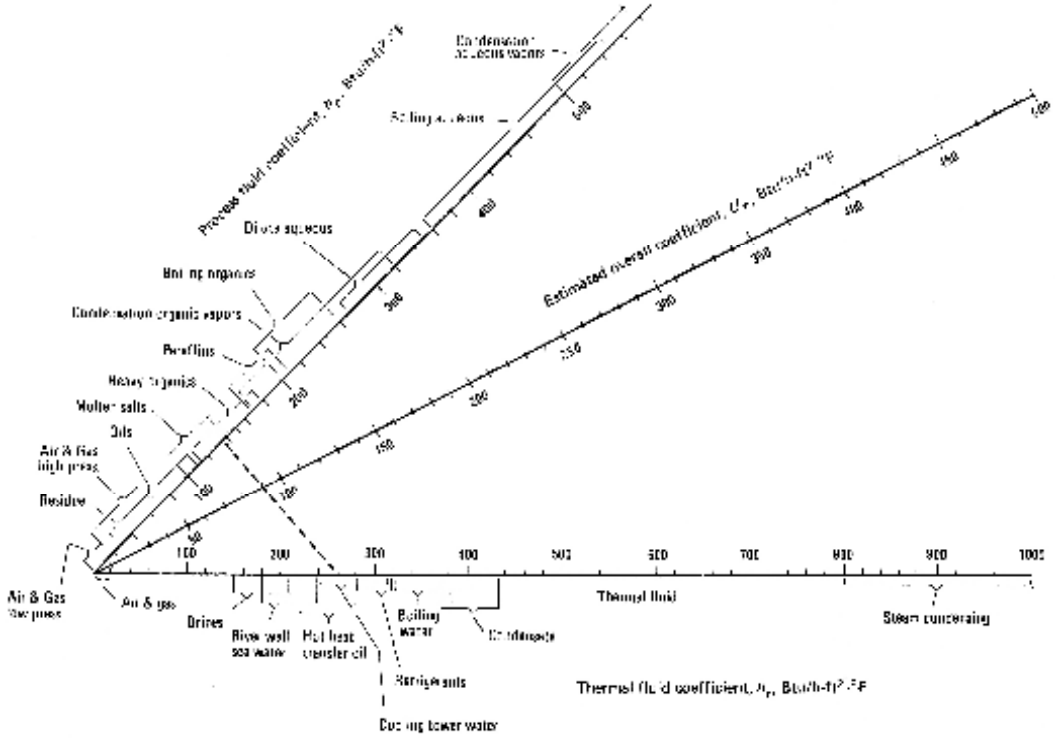


Fig. 2.10

Nomograph for estimating Overall Heat Transfer Coefficients (Ref. 2.10)

The fouling factor (F) in Equ. 2.11 is the only thing that differentiates it from Equ. 2.9. This F is thus an additional resistance in series. Across a given surface there may be fouling on both fluid walls. Thus, Equ. 2.11 could have two values of F in the denominator.

Sometimes a fouling factor is expressed as a number like 1.5, 2.0, 3.0, etc. The actual value of F is found by dividing by 1000, i.e., 0.0015, 0.002, 0.0003, etc.
Table 2.2 Typical Over-all Heat Transfer Coefficients (Ref 2.1)

Typical Over-all Heat Transfer Coefficients—U

SHELL AND TUBE EXCHANGERS

	Coefficient, Btu/Hr-sq.ft.-°F.
Water Coolers	
Gas 10-500 psi	80
Gas over 500 psi	100
C_3, C_3, C_4	90
Natural Gasoline	80
Naphtha	75
Kerosene	70
MEA	145
Air	20
Water	175
Water Condensers	
C_2, C_3, C_4	85
Still Overhead	80
Naphtha	75
Amine Regenerator	110
Reboilers—Steam or Hot Oil	
De C_3, De C_4 Still	130
Glycol Reconcentrator	13
MEA Regenerator	120
Miscellaneous	
Lean Oil/Rich Oil—Naphtha	95
Lean Oil/Rich Oil—Kerosene	80
Lean Oil/Gas	80
Gas/Gas	40
Gas Chiller—Propane Refrigerant	90
Lean Oil Chiller—Propane Refrigerant	95
Lean MEA/Foul MEA	130
Steam/Gas	50

AERIAL COOLERS

	Bare Tube Coefficient	Approximate Air Temperature Rise
Condensers		
Propane, Butane	90	35 °F
Still Overhead	75	35
Steam 0-50 psi	135	70
Naphtha	75	30
MEA Regenerator Overhead	80	60
Natural Gasoline	80	30
Freon	80	35
Coolers		
C_3, C_4	90	30
Natural Gasoline	80	30
Naphtha	70	30
Gas—Low Pressure	50	20
Gas—High Pressure	70	35
Water	120	30
Lube Oil	15	10
Air	35	15

VAPOR-LIQUID EQUILIBRIUM

There are many ways in which equilibrium between vapor and liquid can be expressed mathematically. Perhaps the most widely used expression is shown in Equ. 2.13.

$$y_i = k_i x_i \qquad (2.12)$$

***Where*:**

y_i = mole fraction of one component in the vapor phase
X_i = mole fraction of that same component in the liquid phase
K_i = equilibrium constant

The equilibrium constant is not really a constant in Equ. 2.12. It is a function of temperature, pressure and composition. However, charts and graphs are available for most of the hydrocarbons and natural gas constituents. For many sweetening agents no experimental data are available and resort must be made to estimation techniques. One of these techniques makes use of the fugacity of the component. Using fugacities equation 2.12 can be written as

$$K = \frac{y_i}{x_i} = \frac{f_{i_L}^{\ o}}{f_{i_v}^{\ o}} \qquad (2.13)$$

Where: $f_{i_v}^{o}$ = fugacity of component in the pure vapor state (2.13)

$f_{i_L}^{o}$ = fugacity of component in the pure liquid state

Table 2.3

Fouling Factors

Typical fouling resistances are referred to the surface on which they occur. In the absence of specific data for setting proper resistances the user may be guided by the values tabulated below.

Water

Temperature Of Water	Up to 240°F		240-400°F	
	125°F or less		Over 125°F	
	Water Velocity		Water Velocity	
Types of water	Ft/sec		Ft/sec	
	3 ft. And less	over 3ft .	3ft. And less	over 3ft .
Sea water	0.0005	0.0005	0.001	0.001
Brackish water	0.002	0.001	0.003	0.002
Cooling tower and artificial spary pond treated makeup	0.001	0.001	0.002	0.002
Untreated	0.003	0.003	0.002	0.004
City or well water (such as great lakes)	0.001	0.001	0.005	0.002
Great lakes	0.001	0.001	0.002	0.002
River water minimum	0.002	0.001	0.002	0.002
Muddy or silty	0.003	0.002	0.003	0.003
Hard (over 15 grains /gal)	0.003	0.003	0.004	0.005
Engine jacket	0.001	0.001	0.001	0.001
Distilled	0.0005	0.0005	0.0005	0.0005
Treated boiler feedwater	0.001	0.0005	0.001	0.001
Boiler blowdown	0.002	0.002	0.002	0.002

* Ratings in columns 3 and 4 are based on a temperature of the heating medium of 240-400°F. If the heating medium temperature is over 400°F and the cooling medium is known to scale, these ratings should be modified accordingly.

Liquids			
Industrial organic heat transfer media	0.001	Fuel oil	0.005
Stable side draw and bottom product	0.001	Engine lube oil .	0.001
Molten heat transfer salts	0.0005	Caustic solutions	0.002
MEA &DEA Solutions	0.002	Lean oil	0.001
DEG & TEG solutions	0.002	Rich oil	0.001
Refrigerant liquids	0.001		
Gases and vapour			
Industrial organic heat transfer media	0.001	Overhead products	0.001
Refrigerant vapors (oil bearing)	0.002	manufactured	0.01
Exhaust steam (oil bearing)	0.001	Compressed air	0.002
Stable overhead products . . .	0.001	Solvent vapors	0.001
Steam (non- oil bearing) . . .	0.0005	Natural gas	0.001
Engine exhaust gas	0.01	Acid gas	0.001

(Ref.2.12)

Fugacities can be determined from Figure 2.11 as functions of reduced temperature and reduced pressure.

Equ. 2.12 can be used to predict the composition of the vapor in equilibrium with a liquid of given composition or vice versa. It cannot, however, be used to solve the most frequently encountered process problem - at a given temperature and pressure what proportion of a specified stream will be vapor and what proportion liquid? Based on Equ. 2.12 we can derive an expression for determining the amount of liquid or vapor present.

$$\ell_i = \frac{f_i}{k_i(\frac{V}{L}) + 1} \qquad (2.14)$$

$$L = \sum \ell_i \qquad (2.15)$$

Where:

ℓ_i = moles of one component in the liquid formed

f_i = moles of one component in the feed

V = total moles of vapor

L = total moles of liquid

Equ. 2.14 permits estimation of the amount of liquid formed when a stream of specified composition is flashed at a given temperature and pressure. The corresponding equation for the vapor formed is

$$V_i = \frac{K_i f_i}{\frac{L}{V} + K_i} \qquad (2.16)$$

$$V = \sum V_i \qquad (2.17)$$

Where: V_i = moles of one component in the vapor formed

Equ. 2.14 And 2.16 cannot be solved directly - the solution must be trial and error. We must know the value of. V or L or the ratio of the two in order to solve either of the two equations. However, in the solution of either equation we can calculate the total liquid or vapor formed. The procedure for solution is to assume a value for the total amount of liquid or total amount of vapor formed. Using this as a basis, the liquid or vapor formed is calculated by either Equ. 2.14 or 2.16. When the assumed value and the calculated value are the same, the proper solution has been found.

There are rules which, if followed, will make the solution of Equ. 2.14 and 2.16 simpler and more rapid.

1. Always use the form of the equation that will permit you to calculate the stream that you expect to be present in the,. smallest amount. For example, if you expect only 10% liquid and 90% vapor, you would use Equ. 2.14 And calculate the amount of liquid formed in the flash.
2. Once a calculation has been made, the following rule always applies: If the calculated value differs from the assumed value, the next assumption should be even further in the direction of difference.

Use of these two rules will help cut down the calculation time involved in solving either Equ. 2.14 or 2.16.

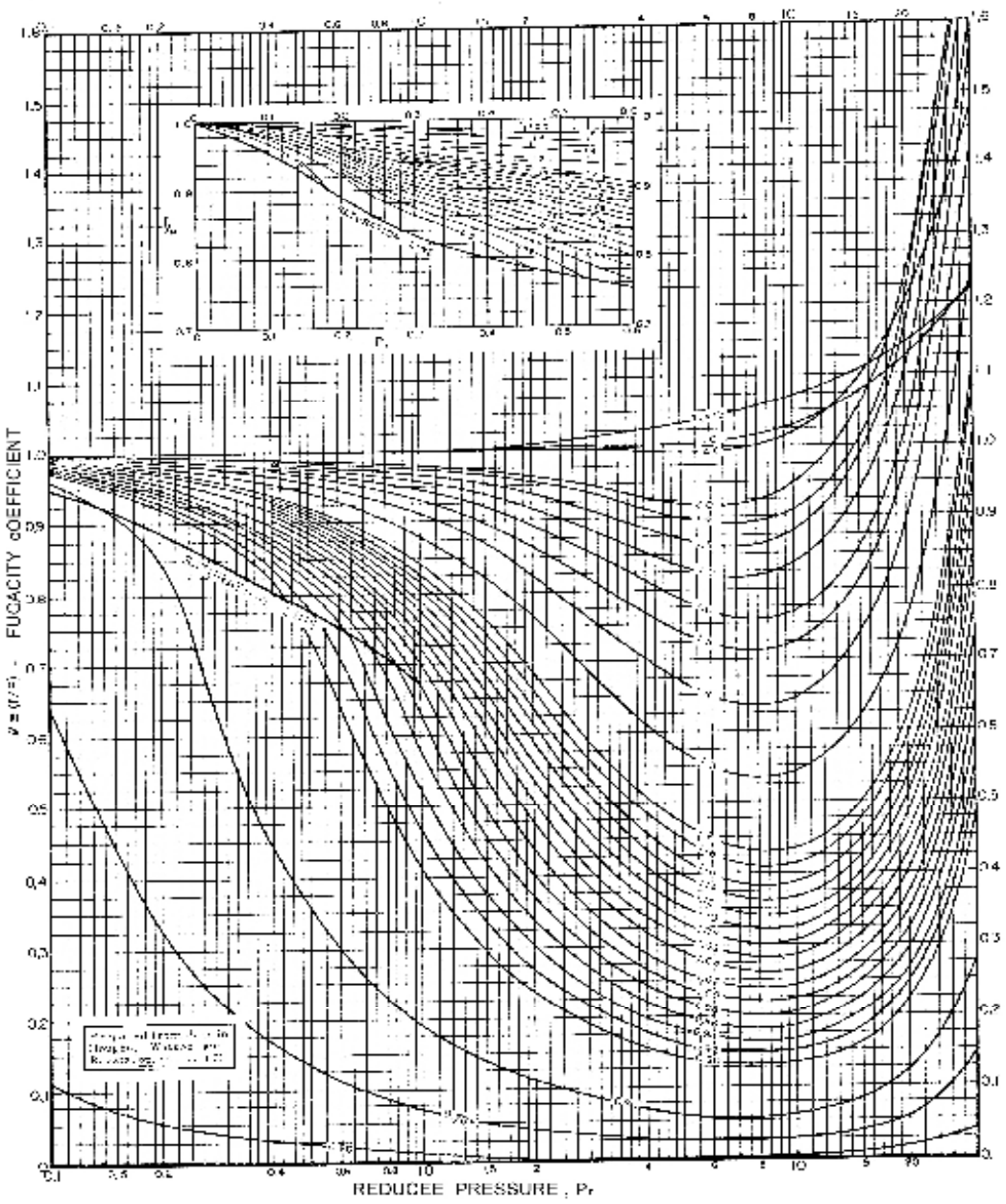


Figure 2.11 (Ref. 2.1)

Generalized Correlation of the Fugacity Function in the Form Fugacity Coefficient (v= f/P) Plot

Example Problem

A stream of the composition shown below is to be flashed to form liquid and vapor under conditions such that the equilibrium constants for the different components are as shown in the table. Calculate the amount of liquid and vapor formed in the flash. Because we expect less vapor to be formed than liquid we will use Equ. 2.16.

Solution

Comp	mol frn	K	Kf	Assume V = 0.014
H_2S	.304	.681	.207	.0029
CO_2	.016	5.28	.0845	.0011
C_1	.024	50.5	1.212	.0100
Solvent	.656	0	0	
	1.000			.014

Since the assumed value for the moles of vapor formed and the calculated value agree, the calculation is correct. As an explanatory note, the solvents used in gas sweetening normally are so non-volatile that they have essentially a zero K value at normal temperatures and pressures.

ABSORPTION AND STRIPPING

In the case of absorption and stripping operations where physical absorp-rion is involved, we can predict solution circulation rates required through the use of the simplified Kremser and Brown type absorption calculation.

$$\frac{Y_{n+1} - Y_1}{Y_{n+1} - Y_0} = \frac{A^{n+1} - A}{A^{n+1} - 1} \qquad (2.18)$$

Where:

Y_{n+1} = moles of a component in the entering rich gas stream per mole of entering rich gas

Y_1 = moles of a component in the dry gas leaving the absorber per mole of entering rich gas

Y_0 = moles of a component in the gas stream that would be in equilibrium with the lean solvent per mole of entering rich gas

n = number of theoretical plates in the column

A = average absorption factor = $\frac{L_O}{KV_{n+1}}$

L_0 = moles of entering lean solvent total moles of entering rich gas

V_{n+1} = total moles of entering rich gas

If the lean solvent is essentially free of a given component, Yo will equal 0 for that component. Equ. 2.18 has been solved and plotted in graphical form for convenience. This graph is shown in Fig. 2.12

In a calculational sense a stripper is simply an up-side down absorber. Trays are numbered from the bottom up with the top plate designated as m. Equ. 2.19 shows the proper form of Equ. 2.18 for use in stripping.

$$\frac{X_{m+1} - X_1}{X_{m+1} - X_0} = \frac{S^{m+1} - S}{S^{m+1} - 1} \qquad (2.19)$$

Where:

S = stripping factor = $\frac{KV}{L}$

X_1 = moles of component in stripped lean oil leaving bottom of the stripper Per mole of rich oil entering the stripper.

X_{m+1} = moles of a component in the rich oil entering the stripper per mole of rich oil entering the stripper

L_{m+1} = moles of rich oil entering the stripper

Y_o = moles of component in stripping medium per mole of stripping medium entering the stripper

V_o = moles of stripping medium entering

X_o = moles of component in liquid in equilibrium with stripping medium per mole of entering rich oil

L_o = moles of lean oil leaving the stripper

Example Problem

We desire to recover 65% of a component from a gas stream. With the solvent used and at the temperature and pressure of the absorber the equilibrium constant (K) for the component is 1.2. How much solvent must be circulated over the absorber?

Solution

We will use Equ. 2.18. If we assume that the solvent is completely free of the component to be

Absorbed, Y_o will be zero and o

$$\frac{Y_{n+1} - Y_1}{Y_{n+1}} = \frac{A^{n+1} - A}{A^{n+1} - 1}$$

In this form, the left hand side of the equation represents the fraction of the component absorbed or, in our case, 0.65. If the absorber contains the equivalent of four theoretical trays we determine from Figure 2.12 that the absorption factor A must be equal to 0.715.

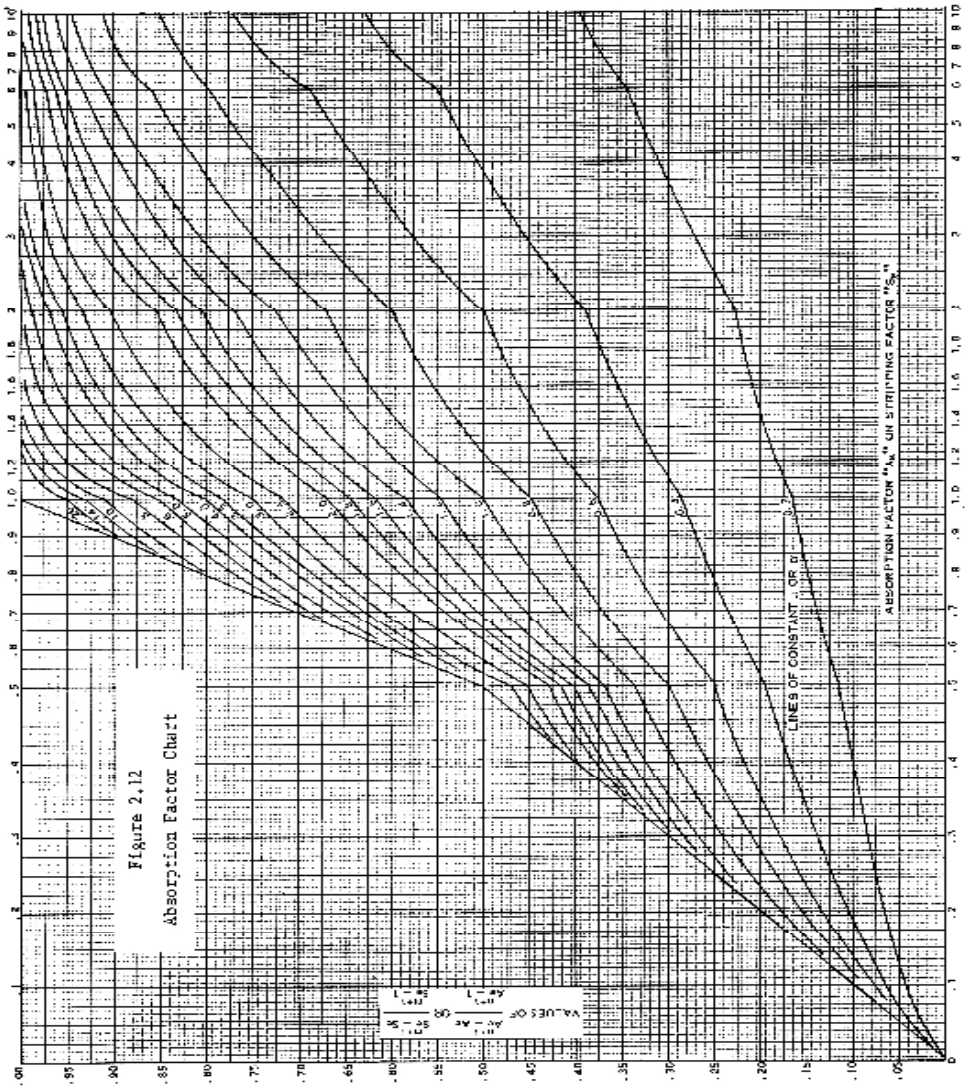


Figure 2.12
Absorption Factor Chart
LINES OF CONSTANT OR
ABSORPTION FACTOR "A" OR STRIPPING FACTOR "S"

Figure 2.12

$$A = \frac{L_O}{KV_{n+1}} \qquad 0.715 = \frac{L_O}{1.2V_{n+1}}$$

$$\frac{L_O}{V_{n+1}} = 0.858 \text{ mols solvent/mol gas}$$

Equ. 2.18 and 2.19 apply to physical absorption and stripping operations. In those cases where a chemical reaction is involved in the sweetening process these two equations may or may not give reasonable approximations to the efficiency of the operation. Predicting rates of chemical reactions is a difficult and involved process. For those cases where chemical reaction is involved the usual procedure is to use either pilot plant data or data on an existing plant as a guide for developing a workable, usable process.

VESSEL SIZING

The sizing of any vapor-liquid separation device, such as an absorber or a separator, is dependent on several factors. The primary driving force for the separation is the difference in density between the liquid and the gas. Equations describing the separation that should occur are relatively easy to formulate from the principles of fluid mechanics and fluid dynamics. There are many practical variables, however, that are difficult to describe, such as particle size distribution, presence of entrained solids, etc. For this reason nearly all techniques for vapor-liquid separation design make use of safety factors based on experience and past performance of vessels that have been designed. A simple but accurate procedure is to use Equ. 2. 20. (2.1)

$$W = C[\ (\rho - \rho_g)\rho_g\ ]^{0.5} \quad (2.20)$$

Where:

W = gas mass velocity pounds/hr/sq ft

C = constant found. From the tabulation below

ρ_ℓ = liquid density in pounds/cubic ft

ρ_ℓ = gas density in pounds/cubic ft

Service	Tray Spacing		
	30"	24"	18"
Absorbers - oil	850	800	700
Absorbers - glycol	550	500	-
Absorbers - amine	395	350	-
Fractionators and Stills	600	540	440

Vertical separators - 600 (8 ft length min)

Horizontal separators - (1440) (L/20)°'56

(Where L = ft between inlet and outlet nozzles)

Example Problem
What would be the maximum allowable gas mass velocity in an amine absorber? Under conditions in the absorber the density of the gas is 3.5 Ib/cu ft and the density of the amine solution is 63 Ib/cu ft. The absorber will have 24 inch tray spacing.

Solution
From the table we see that the value of the constant C for amine absorbers with 24 inch tray spacing is 350.
$W = 350 [(63 - 3.5)]^{0.5} = 350 (208.3)^{0.5} = 5050 \#/ft^2 - hr$

GAS P-V-T BEHAVIOR

There are several ways in which sour natural gases differ in behavior from sweet natural gases. One of these is in terms of the PVT behavior of the gas. The simplest and most direct method for predicting the PVT behavior of natural gases is through the use of the compressibility factor. The compressibility factor is defined through Equ. 2.21

$$PV = ZnRT \qquad (2.21)$$

Where:

P = absolute pressure in psi
V = volume in cu ft
n = number of pound moles of gas
R = universal gas constant = 10.73 ft-lbf/lb mole – ^{o}R
T = absolute temperature °R (°R = °F H- 460)
Z = compressibility factor, dimensionless

The compressibility factor is ordinarily plotted as a function of reduced temperature and reduced pressure according to the theory of corresponding states. A typical plot is shown in Fig. 2.13. The pseudo reduced temperature and pseudo reduced pressure are based on pseudo or molal average critical temperatures and pressures calculated according to Equ. 2.22 And 2.23.

$$P_{pc} = \sum X_i \, p_{c_i} \qquad (2.22)$$

$$T_{PC} = \sum X_I T_{C_I} \qquad (2.23)$$

$$P_{pr} = \frac{p}{p_{PC}} \qquad (2.24)$$

$$T_{Pr} = \frac{T}{T_{PC}} \qquad (2.24)$$

Where:

P_{PC} = pseudo critical pressure psia
x = mole fraction in gas mixture
P = system pressure psia
T_{PC} = pseudo critical temperature, R
T_C = component critical temperature, R

T = operating system temperature, R
P_C = component critical pressure psia
P_{PR} = reduced pressure based on pseudo critical temperature, dimensionless
T_{PR} = system reduced temperature, based on pseudo critical temperature dimensionless
i = subscript = any component in a multicomponent system

After the reduced temperature and reduced pressure have been determined according to Equ. 2.22 to 2.25, the compressibility factor for the gas mixture is read from Fig. 2.13. After the compressibility factor has been determined, the volume of gas can be calculated from Equ. 2.21.

Example Problem
A natural gas of the composition shown below is available at 100 F. and 1000 psia. What is the molar volume and the density of the gas under these conditions?

Comp	MOL Frn	T_C	P_C	W_t	P_{PC}	T_{PC}	Avg Mol wt
C_1	.85	343.4	667.8	16.0	567.6	291.9	13.6
C_2	.12	550.1	707.8	30.1	84.9	66.0	3.6
C_3	.03	666.1	616.3	44.1	18.5	20.0	1.3
					671.0	377.9	18.5

$$T_{Pr} = \frac{560}{377.9} = 1.48 \qquad p_{pr} = \frac{1000}{671} = 1.49$$

From Fig. 2.13, Z = 0.853

Using Equ. 2.21

$$1000 \times V = 0.853 \times 10.73 \times 560 \quad V = 5.13 \text{ ft}/\#\text{mol}$$

$$\rho = \frac{18.5}{5.13} = 3.61 \#/\text{ft}^3$$

The procedure outlined for determining the compressibility factor works very well for sweet natural gas mixtures. Unfortunately, it does not work well for sour natural gases. As the acid gas content of the natural gas increases, the deviation of the natural gas from the compressibility factor behavior shown in Fig. 2.13 increases. A number of efforts have been made to account for this deviation from

Table 2.4
Physical Constants Of Paraffin Hydrocarbons And Other Components Of Natural Gas
NGP publication 2145-71 (1)

Component	Notes	Methane	Ethane	Propane	Iso-Butane	N-Butane	Iso-Pentane	N-Pentane	N-Hexane	N-Heptane	N-Octane	N-Nonane	N-Decane
Molecular Weight		16.043	30.070	44.097	58.124	58.124	72.151	72.151	86.178	100.205	114.232	128.259	142.286
Boiling Point @ 14.696 psia, °F		−258.69	−127.48	−43.67	10.90	31.10	82.12	96.92	155.72	209.17	258.22	303.47	345.48
Freezing Point @ 14.696 psia, °F		−296.46[d]	−297.89[d]	−305.84[d]	−255.29	−217.05	−255.83	−201.51	−130.59	−131.05	− 70.16	− 64.28	− 21.36
Vapor Pressure @ 100°F, psia		(5000)	(1000)	190	72.2	51.6	20.44	15.570	4.956	1.620	0.537	0.179	0.0597
Density of Liquid @ 60°F & 14.696 psia													
Specific Gravity @ 60°F/60°F	a,b	0.3[c]	0.3564[b]	0.5077[b]	0.5631[b]	0.5844[b]	0.6247	0.6310	0.6640	0.6882	0.7068	0.7217	0.7342
°API	a,b	340[c]	265.5[b]	147.2[b]	119.0[b]	110.6[b]	98.0	92.7	81.0	74.1	68.7	64.6	61.2
Lb/gal @ 60°F, wt in vacuum	c	2.5[c]	2.972[b]	4.233[b]	4.695[b]	4.872[b]	5.208	5.261	5.536	5.736	5.893	6.017	6.121
Lb/gal @ 60°F, wt in air	c	2.5[c]	2.962[b]	4.223[b]	4.686[b]	4.863[b]	5.198	5.251	5.526	5.728	5.883	6.008	6.112
Density of Gas @ 60°F & 14.696 psia													
Specific Gravity, Air = 1.00, ideal gas		0.5539	1.0382	1.5226	2.0068	2.0068	2.4911	2.4911	2.9753	3.4596	3.9439	4.4282	4.9125
Lb/M cu ft, ideal gas		42.38	79.24	116.20	153.16	153.16	190.13	190.13	227.09	264.05	301.01	337.98	374.94
Volume Ratio @ 60°F and 14.696 psia													
Gal/lb mol		6.4[c]	10.12[b]	10.42[b]	12.38[b]	11.93[b]	13.85	13.71	15.57	17.46	19.39	21.32	23.24
Cu ft gas/gal liquid, ideal gas		59[c]	37.5[b]	36.43[b]	30.65[b]	31.81[b]	27.39	27.67	24.38	21.78	19.58	17.80	16.30
Gas vol/liquid vol, ideal gas		443[c]	240.5[b]	272.5[b]	220.0[b]	237.9[b]	204.93	207.00	182.37	163.50	146.45	133.18	122.13
Critical Conditions													
Temperature, °F		−116.63	90.09	206.01	274.98	305.65	369.10	385.7	453.7	512.8	564.22	610.68	652.1
Pressure, psia		667.8	707.8	616.3	529.1	550.7	490.4	488.6	436.9	396.8	360.6	332	304
Gross Heat of Combustion @ 60°F	p												
Btu/lb liquid		—	22,214[d]	21,513[d]	21,091[d]	21,199[d]	20,889	20,928	20,784	20,681	20,604	20,544	20,494
Btu/lb gas		23,885	22,323	21,665	21,237	21,290	21,040	21,080	20,944	20,840	20,762	20,701	20,649
Btu/cu ft, ideal gas		1009.7	1768.8	2517.5	3252.7	3262.1	4000.3	4009.6	4756.2	5502.8	6249.7	6996.5	7742.1
Btu/gal liquid		—	65,956[d]	91,065[d]	99,032[d]	102,959[d]	106,790	110,102	115,060	118,668	121,419	123,613	125,444
Cu ft air to burn 1 cu ft gas - ideal gas		9.54	16.70	23.86	31.02	31.02	38.18	38.18	45.34	52.50	59.65	66.81	73.97
Flammability Limits @ 100°F & 14.696 psia													
Lower, vol % in air		5.0	2.9	2.1	1.8	1.8	1.4	1.4	1.2	1.0	0.96	0.87*	0.78*
Upper, vol % in air		15.0	13.0	9.5	8.4	8.4	(8.3)	8.3	7.7	7.0	–	3.9	2.6
Heat of Vaporization @ 14.696 psia													
Btu/lb @ boiling point		219.22	210.41	183.05	157.53	165.65	147.13	153.59	143.95	136.01	129.53	123.76	118.68
Specific Heat @ 60°F & 14.696 psia													
C_p gas — Btu/lb, °F, ideal gas		0.5266	0.4091	0.3881	0.3872	0.3827	0.3827	0.3803	0.3804	0.3875	(0.3876)	0.3849	0.3835
C_p gas — Btu/lb, °F, ideal gas	*	0.4027	0.3436	0.3430	0.3530	0.3525	0.3552	0.3603	0.3653	0.3677	0.3702	0.3685	0.3695
$N = C_p/C_v$	*	1.303	1.193	1.131	1.097	1.097	1.078	1.056	1.061	1.054	1.047	1.042	1.038
C_p liquid — Btu/lb, °F	*	—	0.6256	0.5020	0.5615	0.5636	0.5353	0.5441	0.5332	0.5283	0.5246	0.5228	0.5208
Octane Number													
Motor clear		—	+ 95[f]	97.1	97.6	89.6[f]	90.3	62.6[f]	26.0	0.0	–	–	–
Research clear		—	+146[f]	+1.8[f]	+0.10[f]	93.8[f]	92.3	61.7[f]	24.8	0.0	–	–	–
Refractive Index n_D @ 68°F		–	–	–	–	1.3326[b]	1.35373	1.35748	1.37486	1.38764	1.39743	1.40542	1.41189

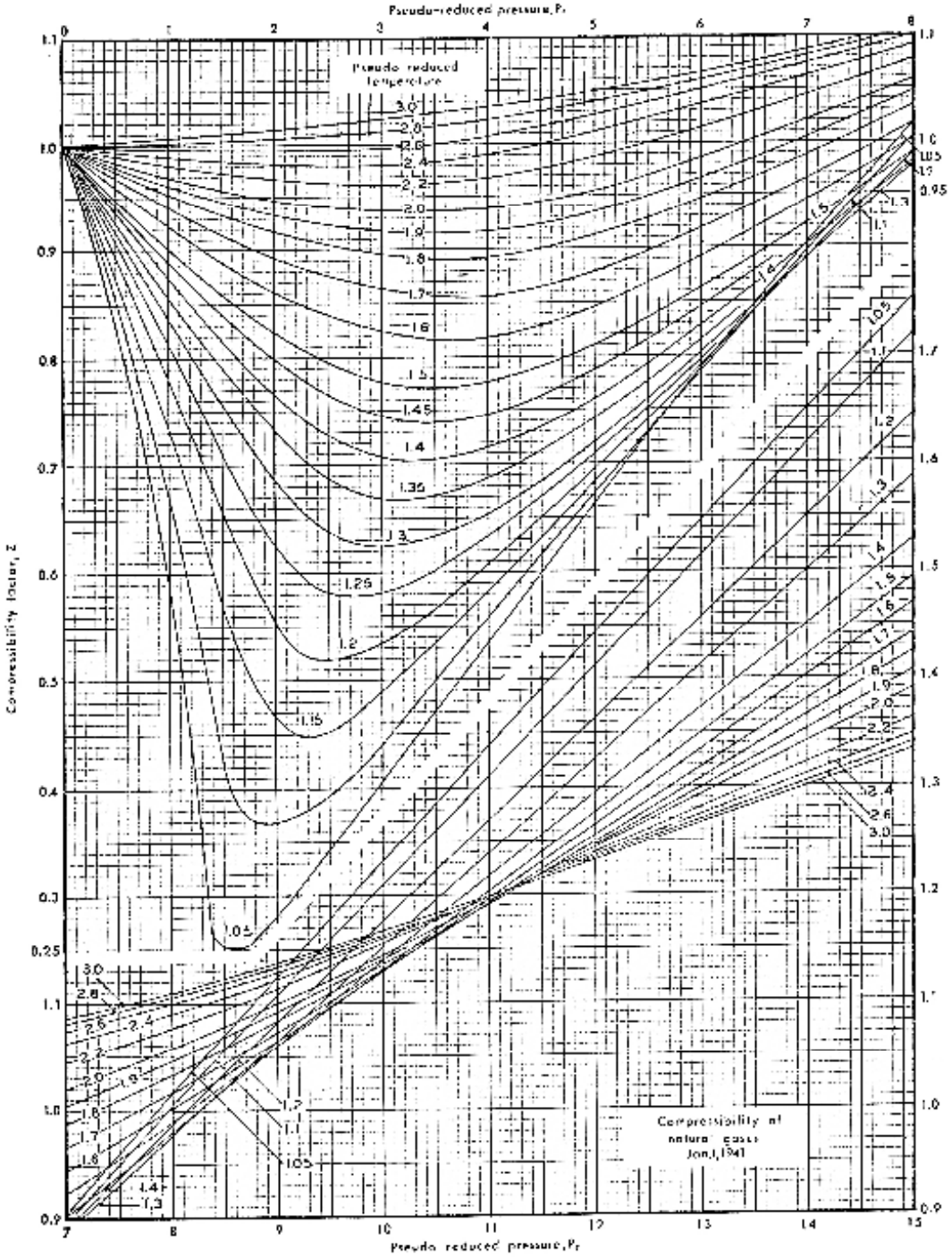
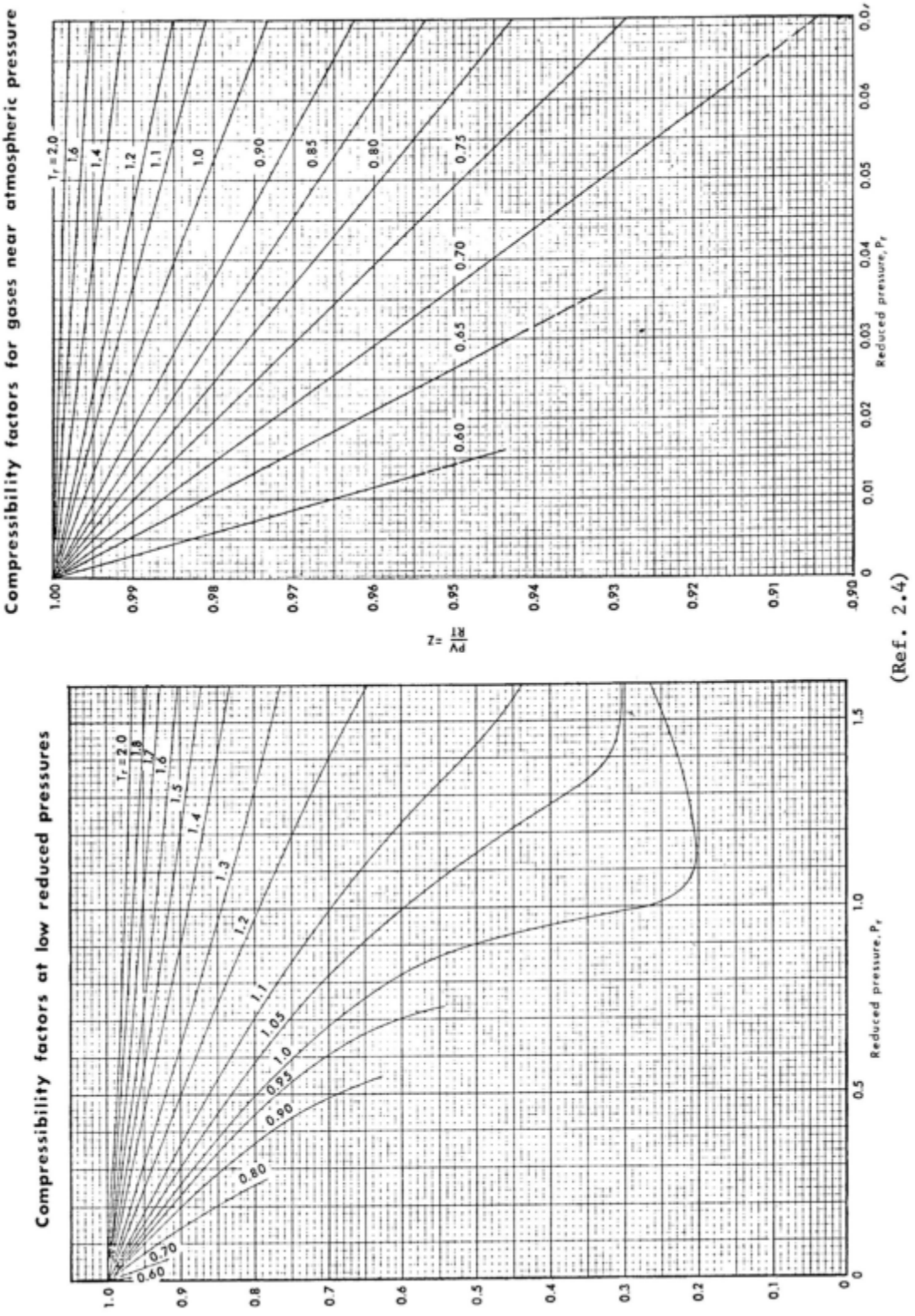


Figure 2.13
Compressibility factors for natural gas


Compressibility factors for gases near atmospheric pressure
Compressibility factors at low reduced pressures
Reduced pressure, P_r
$Z = \frac{PV}{RT}$
(Ref. 2.4)

Sweet gas behavior. Perhaps the simplest, and among the most successful, is that presented by Wichert and Aziz. (2.3) They recommend the use of a pseudo critical temperature adjustment factor which is a function of total acid gas composition and the H_2S to Co_2 ratio in the acid gas. A plot of the pseudo critical temperature adjustment factor as a function of C02 and H_2S composition is shown in Fig. 2.14
The procedure recommended by Wichert and Aziz is as follows:

1. Calculate the pseudo critical temperature and pressure of the gas mixture by Equ. 2.22 And 2.23.
2. From the known concentrations of C02 and H2S in the mixture, determine the pseudo critical temperature adjustment factor from Fig. 2.14.
3. Calculate the adjusted pseudo critical temperature as:

$$T`_{PC} = T_{PC} - \varepsilon \qquad (2.26)$$

Where:

$T`_{PC}$ = the adjusted pseudo critical temperature, °R

ε = pseudo critical temperature adjustment factor as determined from gas composition and Fig. 2.14

4. Calculate the adjusted pseudo critical pressure according to:

$$P`_{PC} = \frac{P_{PC} \times T'_{PC}}{T_{PC} + B(1-B)\varepsilon} \qquad (2.27)$$

Where:

P`= adjusted pseudo critical pressure psia

B = mole fraction of I^S in the sour gas

5. Calculate the pseudo reduced pressure and the pseudo reduced temperature according to Equ. 2.24 and 2.25, but using the adjusted pseudo critical values calculated in steps 3 and 4.
6. Obtain the compressibility factor from Fig. 2.13 using the pseudo reduced pressure and temperature values calculated in step 5.

Wichert and Aziz tested their procedure over a wide range of temperatures, pressures and acid gas compositions. They found it to do an excellent job of reproducing experimental PVT measurements for sour gas mixtures.

Example Problem
What would be the correction factors to be applied to the critical temperature and critical pressure of a natural gas that contains 15% CO and 20% H S?

Solution
From Fig. 2.14, ε = 31° F.
The correction factor for the pseudo critical pressure requires knowledge of both the pseudo critical temperature and the pseudo critical pressure before the calculation can be made. However, having determined e from Fig. 2.14 makes the calculation by Equ. 2.27 simple.

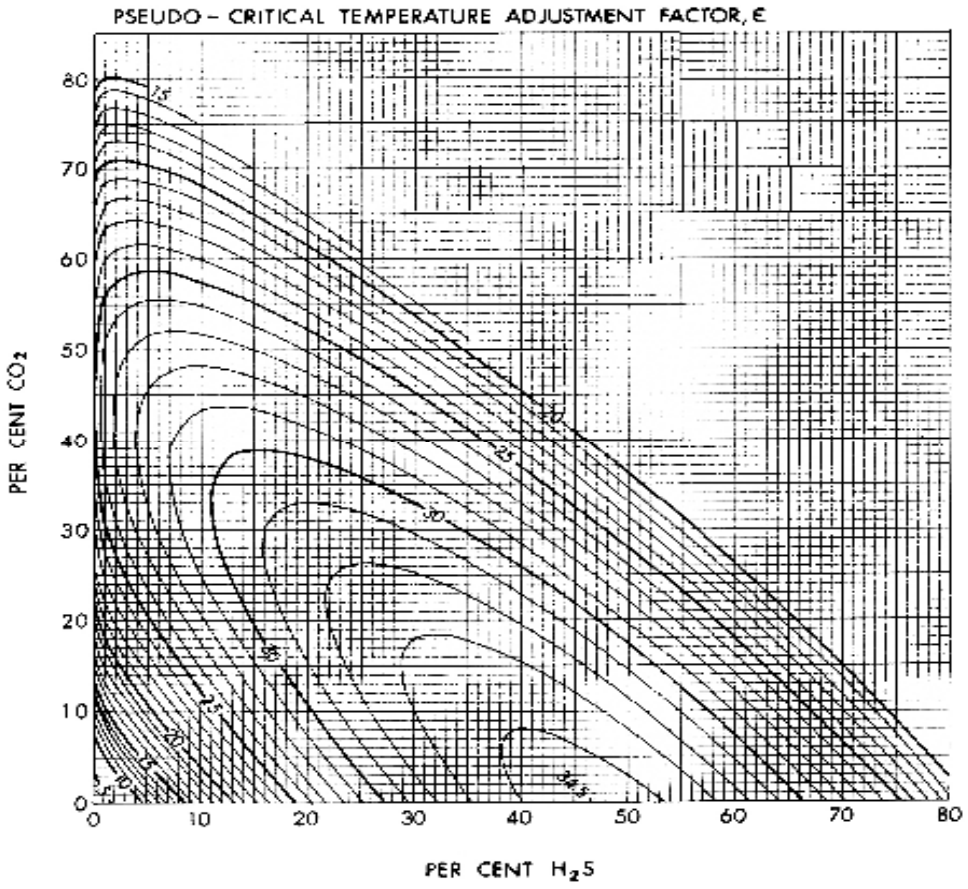


Fig. 2.14
Pseudo critical temperature adjustment factor, e, JR (Ref. 2.3. Courtesy Hydrocarbon Processing)

WATER CONTENT

Prediction of the water content of sour gases is more difficult than for sweet gases, Both H_2S And $C0_2$ are Ver7 soluble in water and possess polar type molecular behavior (like water).

The most precise approach now available utilizes compressibility factor, reduced temperature and reduced pressure calculated using the EMR approach.(2.1) The calculation may be made manually but is tedious.

An alternative, less precise, approach is convenient and produces satisfactory results. It is based on a new correlation of sour gas data.(2.8)
The equation for approximating water content is

$$W = y^{w}_{hc} + y_1 W_1 + y_2 W_2 \qquad (2.28)$$

Where:

- W_{hC} = water content of gas in Ibs water per MMscf/D measured at 14.7 psia and 60° F. (Fig. 2.15)
- W_1 = effective water content contribution of H_2S to the gas mixture (Fig. 2.16)
- W_2 = effective water content contribution of CC>2 to the gas mixture (Fig. 2.17)
- y = mole fr. of all components in gas except H_2S and $C0_2$
- y_1 = mole fr. of H_2S in gas
- y_2 = mole fr. of CO_2 in gas

Example: Calculate the water content of a natural gas at 100 F. and 1000 psia containing 9.5% H_2S and 2.1% CO_2

From Fig. 2.15, 2.16, and 2.17, W_{hC} = 61, W_1 =150 and W_2 = 68
W = (0.884) (61) + (0.095) (150) + (0.021) (68)
=69.6 Ibs water/MMscf/D

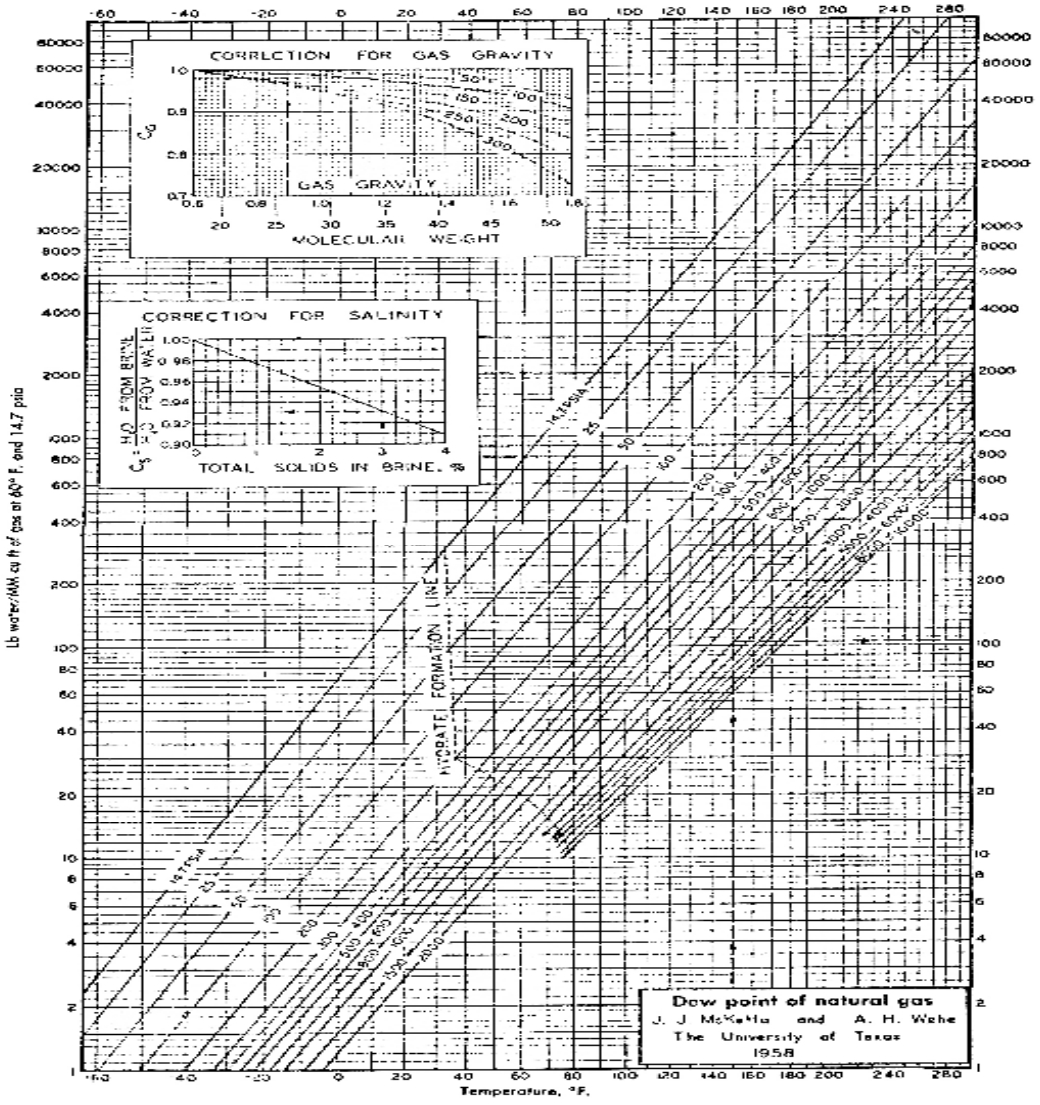
Both Fig. 2.16 and 2.17 are based on data for pure components, and mixtures containing up to 30% E^S and/or CO_2. The water content shown is an effective value to make Equ. 2.28 fit actual water content data. All data are for systems where water is the only liquid phase; there is no liquid H_2S or C02 phase present.

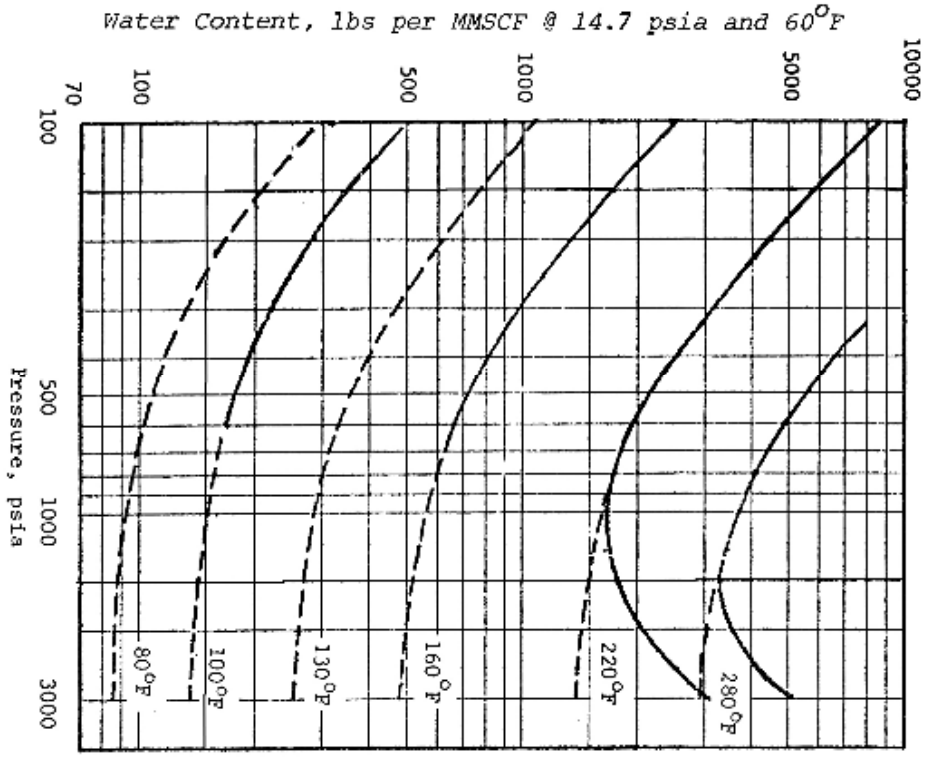
In Fig. 2.16 the solid lines are based on pure H_2S data, the dashed lines on mixture data. The solid lines end at the dew point conditions for pure H2S. Above 220° F. the dew point pressure is above 5000 psia. Two lines are thus shown, a solid one reported for pure H2S and a dashed line for expected mixture behavior.

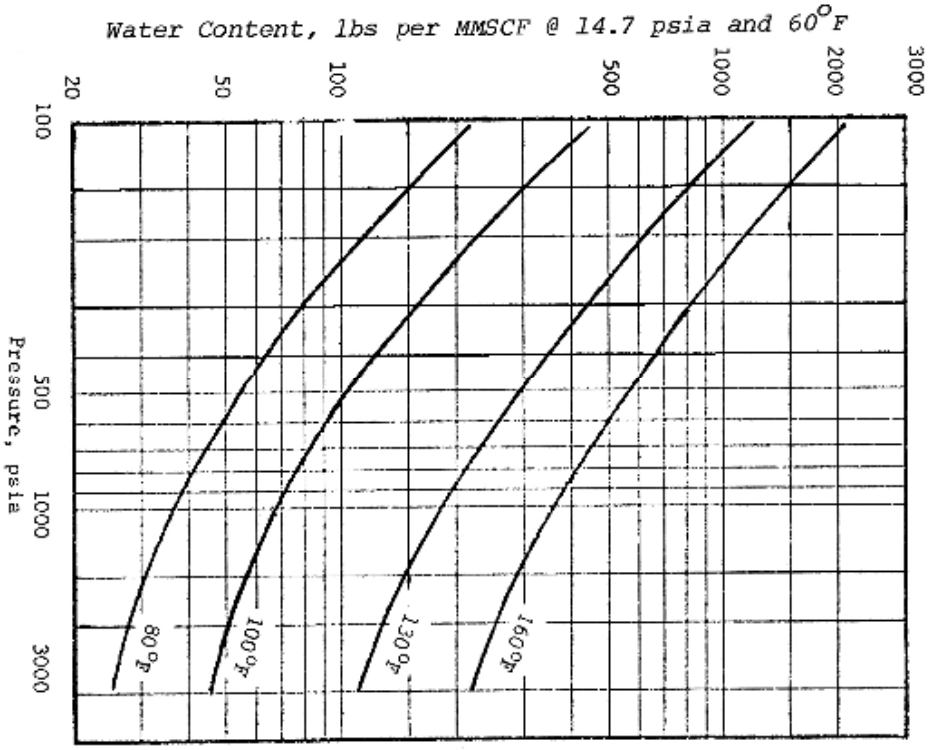
Below300-psia the water content of pure saturated H2S and CO_2 is equivalent to natural gas. As pressure increases, the water content of sour gases increases above sweet gas.

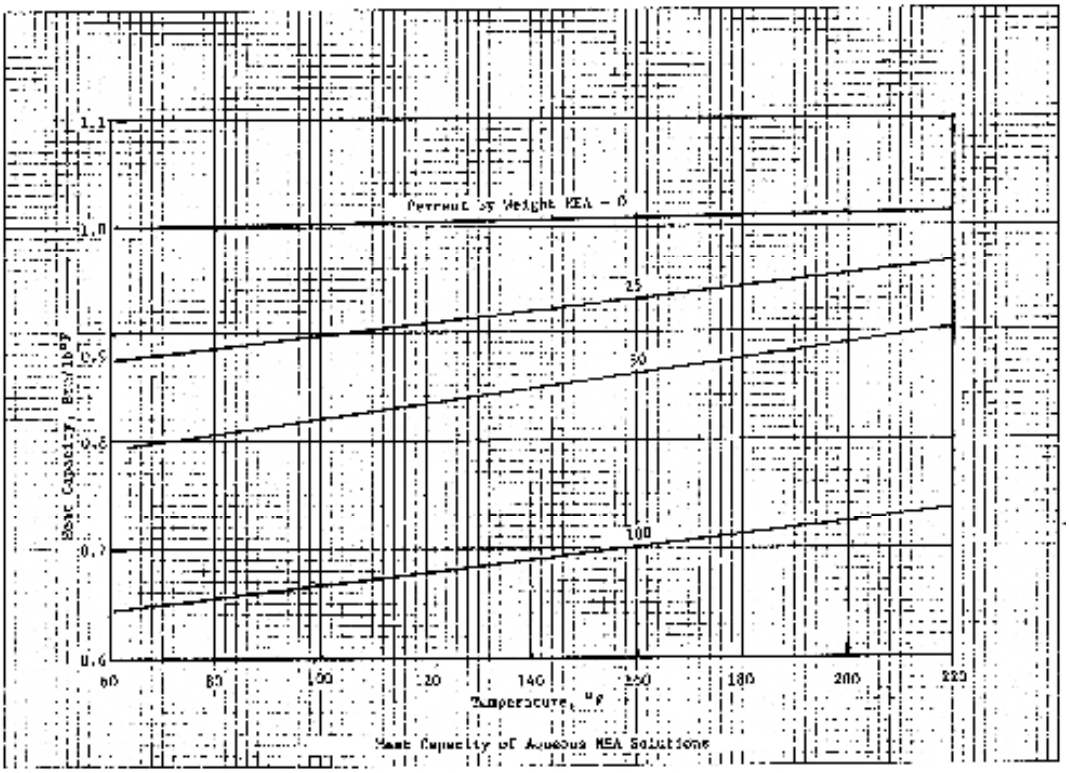
From our experience, Equation 2.28 will give the maximum value of the water content expected for the gas. The actual water content likely will be lower than that predicted. For further details see "Gas Conditioning and Processing", a companion book in the Campbell Petroleum Series.

Remember: the water content shown in the figures and equations is for a sa tura ted_gas. At its water dew point.


CORRECTION FOR GAS GRAVITY
GAS GRAVITY
MOLECULAR WEIGHT
CORRECTION FOR SALINITY
TOTAL SOLIDS IN BRINE, %
HYDRATE FORMATION LINE
Dew point of natural gas
J. J. McKetta and A. H. Wehe
The University of Texas
1958
Temperature, °F.
Lb water/MM cu ft of gas at 60° F. and 14.7 psia


Water Content, lbs per MMSCF @ 14.7 psia and 60°F
70
100
500
1000
5000
10000
100
500
1000
3000
Pressure, psia
80°F
100°F
130°F
160°F
220°F
280°F


Water Content, lbs per MMSCF @ 14.7 psia and 60°F
20
50
100
500
1000
2000
3000
100
500
1000
3000
Pressure, psia
80°F
100°F
130°F
160°F


Percent by Weight MEA - 0
25
50
100
1.1
1.0
0.9
0.8
0.7
0.6
60
80
100
120
140
160
180
200
220
Temperature, °F
Heat Capacity of Aqueous MEA Solutions

MODULE 2:

COMPRESSOR

COMPRESSOR

INTRODUCTION

Compression is used in all aspects of gas processing such as gas lift, gas gathering, helium recovery, condensate recovery, transmission and distribution, re-injection for pressure maintenance, gas storage, and liquefaction for transport.

Figure 10-1 classifies the various types of compressors by their method of operation. However, in the "oil patch" the most common types are reciprocating, centrifugal, sliding vane, and rotary screw.

Reciprocating compressors are widely used in the process industry because they are flexible in throughput and discharge pressure range. Driver horsepower requirements vary from less than 1 to 10000 hp or higher. Speeds may range from 125 to 1000 rpm. Piston speeds range from 500 to 950 ft/min, the majority being 700 to 850 ft/min. Nominal gas velocities usually range from 4500 to 8000 ft/min, and discharge pressures vary from vacuum to 50000 psig (James, 1982b).

Sliding vane compressors are limited to low pressure (up to 130 psig), small flows (50 to 3500 ACFM), and clean gases. Their advantages include low cost, high compression and overall efficiencies, high volumetric efficiencies due to low slip, and relatively low maintenance. They are often used for vapor recovery from stock tanks.

Rotary screw compressors are used for air compression and vapor recovery units and have even been used in refrigeration units. Where the flood-oil/gas separation can be accomplished satisfactorily rotary-screw compressors are very attractive. Rotary screw compressors are relatively recent and promising "newcomers" to the oilfield.

Centrifugal compressors are used in a wide variety of applications in chemical plants, refineries, and in field gathering of natural gas. Centrifugal compressors can be used for outlet pressures as high as 10000 psia, thus overlapping with reciprocating compressors over a portion of the flow-rate/pressure domain. Centrifugal compressors

Have a definite lower limit of flow rate, however, due to the lower limit in physical size of the impeller wheels. Flow rates of 100 to 200000 cfm at inlet conditions can be handled in these devices. The lower limit of 100 cfm is for any impeller, which is usually the final stage. Mass flow rate depends on the inlet temperature and pressure, of course.

Every compressor requires a source of energy or driver or prime mover. The more common types are steam turbines, electric motors, gas turbines, and gas engines. Sometimes diesel engines are used, but mainly as emergency drivers.

Historically the internal-combustion gas engine has been the leading prime mover in the oil patch. It has maintained its popularity in the face of strong challenge by the gas turbine because of its overall efficiency and the ready availability of natural gas. In addition, gas engines and reciprocating compressors make compact integral units in sizes from 200 to over 7000 HP. Large gas-engine/ reciprocating-compressors can achieve 40% overall thermal efficiency with extensive maintenance. Gas plants with smaller units can expect about 25%. Very large steam turbines with, say, five stages of regenerative feed heating can achieve 33% thermal efficiency. Thermal efficiencies of 20 to 25% are typical of the simpler cycles used in the field. Smaller size simple-cycle gas-turbine/centrifugal-compressor units usually achieve 25% thermal efficiency. This comparison allows no credit for the heat content of the low-pressure steam or the gas turbine exhaust gases.

For about 40 years, gas turbines have proved to be successful prime movers in oil fields, refineries and process industries. In fact, gas turbines provide high power outputs at high speeds need only limited space, are light weight, have good reliability and low maintenance, and offer 1000 to 100000+ HP sizes.

Steam turbines are most desirable where there is demand for low-pressure steam for process and/or heating, e.g., refineries and petrochemical plants. Electric motors provide convenience, efficient operation, low maintenance, and
Natural Gas Compression

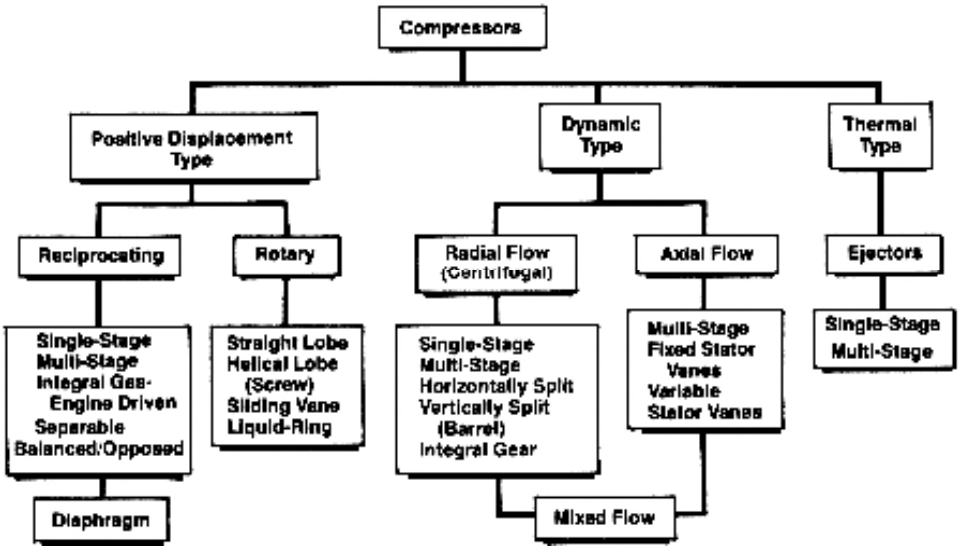


Figure 9-1
Types of compressors (GPSA, 1987, p. 13-2)

Design flexibility to refineries, petrochemicals, and gas processing plants. Because these two types are not the most common drivers for oilfield compressors they are not described here; excellent reviews are available, e.g., GPSA (1987, Section 15); Bloch et al. (1982), or Neerken (1980). Oilfield pumps and their most common drivers—electric motors—are discussed in Volume 2.

A very important feature of any compressor installation is the suction scrubber. It is necessary to prevent contaminants—solids or liquids—from entering a compressor. Solids are often present in gas streams, either picked up in previous piping and equipment or rust and mill scale from the pipe and equipment walls. Salt may accumulate as brine evaporates. Liquid droplets result from condensation of water and/or process condensate or entrained corrosion inhibitors, glycol, or amine. Reciprocating compressors are particularly susceptible to dirt and liquid entry because the latter destroy cylinder oil film. Centrifugal compressors can tolerate some liquid entry in mist form but not in slug form. In any event, good practice demands an adequately sized suction scrubber or inlet vapor-liquid separator. Vapor-liquid separators are discussed in Volume 2.
Gas compression is reviewed using the following topics: reciprocating compressors, rotary compressors, centrifugal compressors, compressor design, gas engines, gas turbines, preventive maintenance, and selection of compressors and drivers.

RECIPROCATING COMPRESSORS

Compression Cycle

The main elements of a reciprocating compressor are the piston and cylinder. The driver moves the piston inside the cylinder, alternately pulling in and pushing out the process fluid (Fig. 10-2). The circular motion of the crank end of the connecting rod is translated into linear back-and-forth motion by the wrist-pin connection between the connecting and piston rods, guided by the crosshead sliding in its bearing.

As illustrated in Figure 10-3, the reciprocating cycle consists of four steps: compression, discharge, expansion, and suction. In Figure 10-3, compression starts at point 1 with the piston all the way out (bottom dead center) and both the suction and discharge valves closed. During compression (points 1 to 2), the piston moves in and compresses the gas trapped in the cylinder. At point 2 the discharge valve (D) opens, and the compressed gas is discharged between points 2 and 3. When the piston is all the way in at point 3, the discharge valve closes and the trapped gas expands from discharge to suction pressure as the piston moves back to point 4. At point 4 the suction valve (S) opens, and gas is drawn into the cylinder as the piston continues moving back to point 1, thus completing the cycle. Figure 10-3 describes the theoretical cycle based on the standard assumptions of isentropic compression and expansion and zero pressure differential in the valves. An actual cycle, shown in Figure 10-4, is obtained using a Beta analyzer

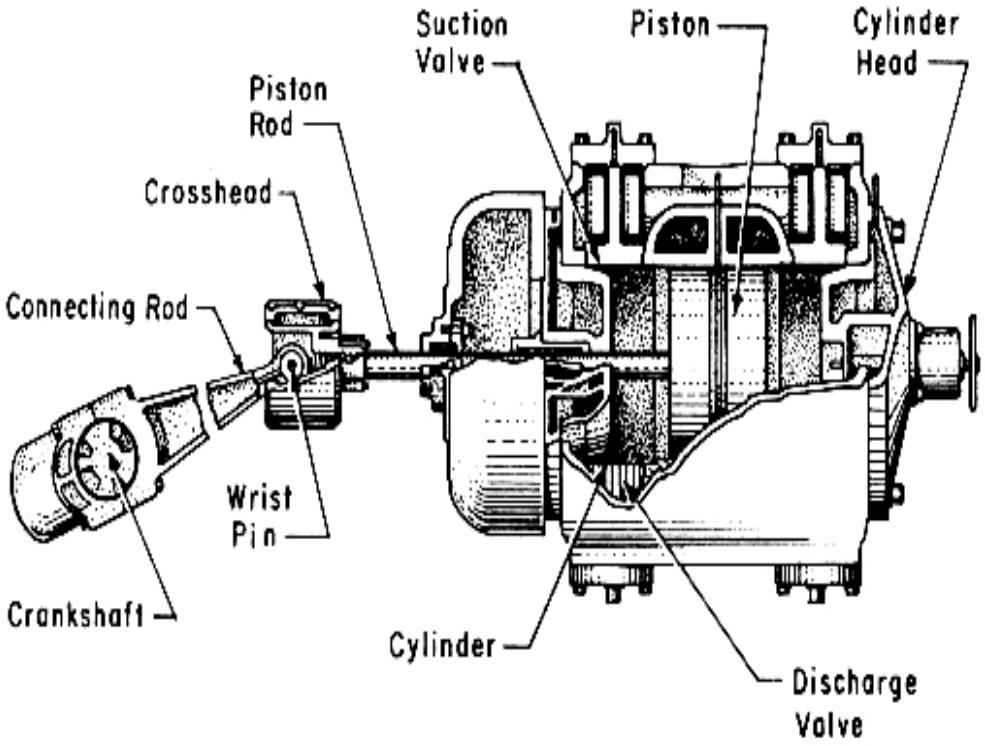


Figure 9-2

Components of a reciprocating compressor (courtesy PETEX, 1987. Copyright © 1987 by the University of Texas at Austin).

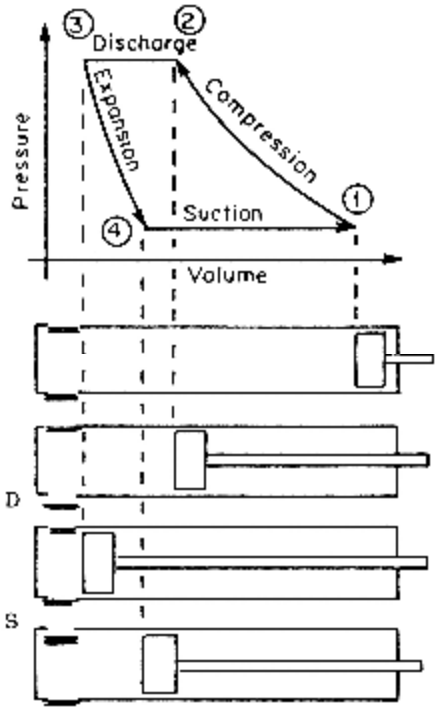


Figure 10-3
Ideal P-V diagram for reciprocating compressor (reprinted from James, 1982b, courtesy of Marcel Dekker Inc.).

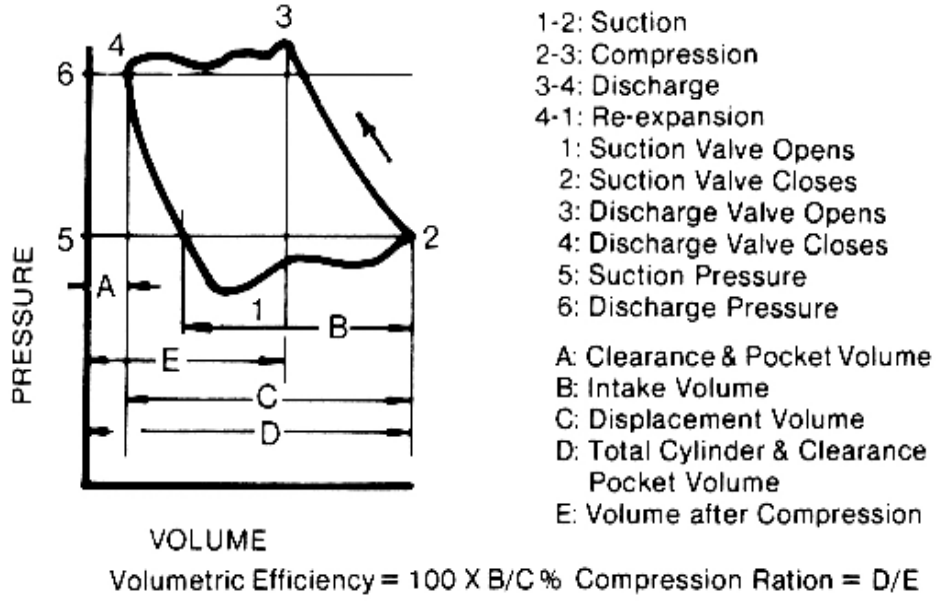


Figure 10-4
Actual P-V diagram for reciprocating compressor (PMC/Beta Corp.).

Reciprocating compressors are prone to wear and tear during operation. The great number of moving parts, the sliding friction, the reversal of direction at rapid intervals, and the mechanical shock involved all contribute to severe wear.

A double-acting cylinder balances some of the load on the piston and rod components. Now both sides of the cylinder are used to compress gas, one side being in suction when the other is in compression. Double-acting pistons are often used for the first stages of compression while single-acting are common in the higher stages.

Valves

A reciprocating compressor resembles an internal combustion piston engine in construction and operation. One very important difference is the kind of valve used. Piston engines use tulip valves that are lifted by rods actuated by a camshaft. Piston compressors use automatic valves that operate from the pressure differential between line and cylinder pressure. On the suction stroke, the pressure in the cylinder must fall slightly below that of the gas entering the cylinder. Likewise, upon discharge the pressure in the cylinder must be slightly higher than that in the discharge line in order to actuate the valve. The flow efficiency of these valves is, therefore, paramount for overall compression efficiency.

Depending on the rotating speed, gas compressed and operating pressures, different valve concepts with various geometries are used (Fig. 10-5). Plate and ring valves prevail in medium and high speed gas compressors. Channel and strip valves are used in air compressors. Poppet valves are found in gas transmission applications and low pressure process compression equipment.

Non-interchangeability between Suction & Discharge
Springs color coded for proper identification
Patented flow path minimizes valve loss HP
Mass damping plate/spring system
Dehnebolt design meets API 61B
Patented scallop valve plate

Figure 10-5a.
Plate valve, Type PF (Dresser-Rand, 1990).

Figure 10-5b
Plate valve, Type CT (Hoerbiger, 1990).

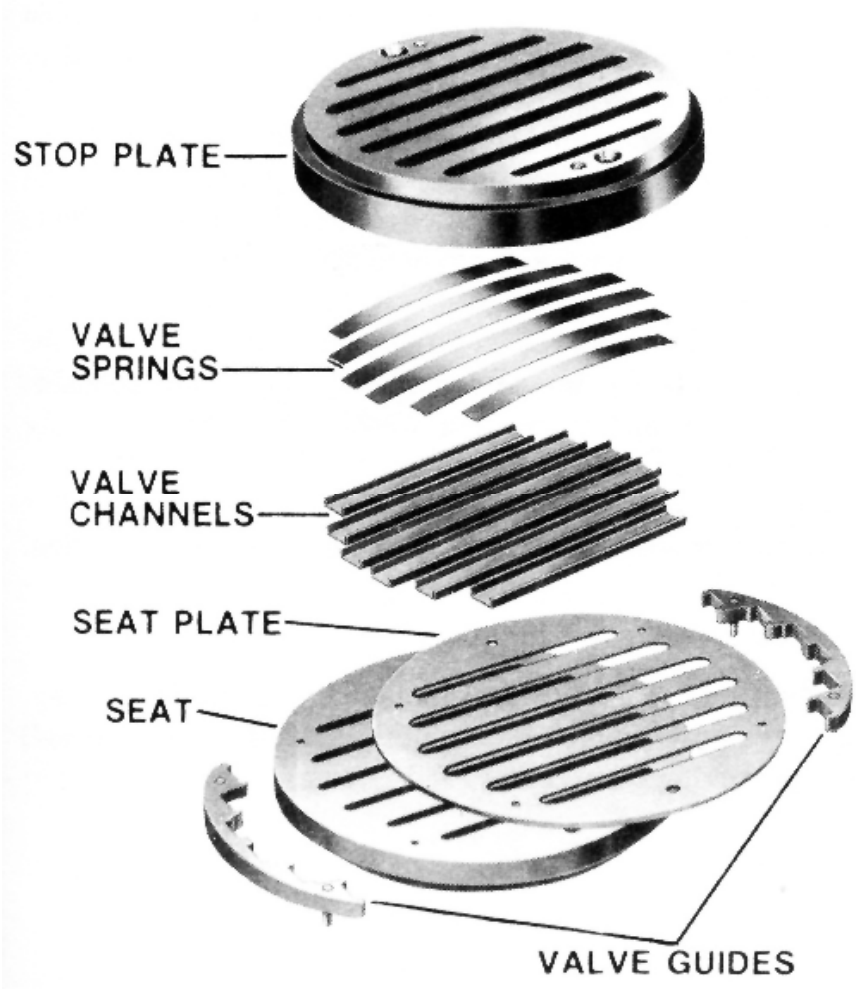


Figure 10-5b
Channel valve (Dresser-Rand, 1990).

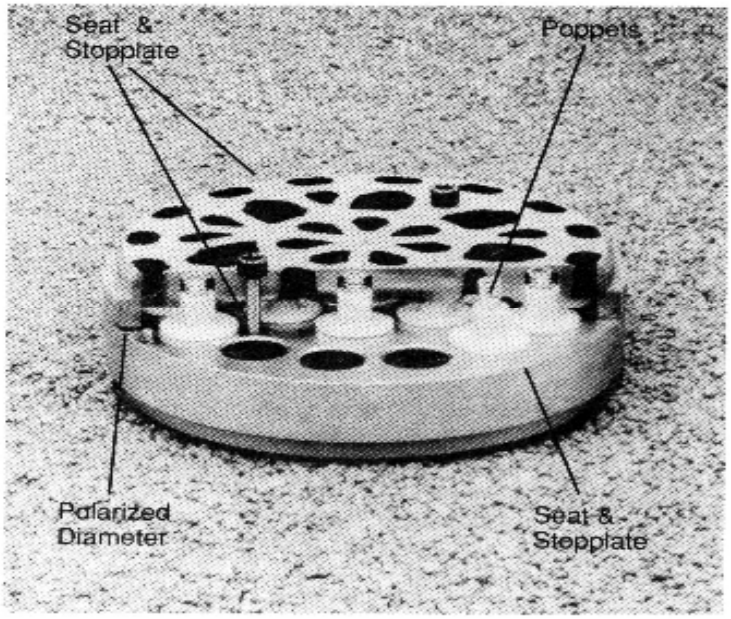


Figure 10-5b
Poppet valve (Dresser-Rand, 1990).

Compressor valves need to be tailored to the operating conditions for optimum performance and valve life. The proper selection of springing is important when designing a valve for the given parameters of operation.

Although valves are considered maintenance items, a well-designed compressor valve should achieve a life cycle of uninterrupted service, depending on operating speed, usually from six months and extending to several years in slow speed and clean environment.

Clearance

Compressor throughput can be controlled by regulating the, speed of the prime mover. The number of strokes per minute times the displacement volume gives the inlet (or suction) flow rate of gas delivered. When feasible, varying prime-mover speed is usually the best and simplest way to control throughput. When prime-mover speed cannot be varied sufficiently, the clearance in the head end of the cylinder is changed. Two types of clearance control are used (Fig. 10-6). In the first, a fixed change in clearance is obtained, either by lifting a plug using a hand-operated crank (Fig. 10-6a) or by actuating a relay-operated valve or alternatively by changing out a plug in the head end during shutdown. In the second method, variable clearance is provided by a movable piston mounted in the head and adjusted using a crank-driven, threaded shaft (Fig. 10-6b). The larger the clearance volume at top dead center, the more slowly the pressure rises during the compression stroke and falls during the suction stroke—and the smaller the volume of intake gas. Figure 10-7 shows the reduction in throughput achieved by increasing clearance.

Unloaders

Sometimes the flow through a compressor must be shut off faster than can be done by shutting down the prime mover This is accomplished using an unloader, .which causes intake valve stay open throughout the stroke; no gas can be delivered because the pressure cannot increase above the suction pressure. Unloaders are often required for compressor start up to reduce starting torque. Figure (10-6b) depicts a hand-operated valve-lifter or unloader.

Adjusting clearance and unloading clearly waste energy compared to slowing down or shutting off the driver, because the driver is not as efficient as at normal design conditions. This inefficiency is important when operating installations where several different compressor combinations can supply the required gas flow. Some combinations will be more energy efficient than others.

Pulsation Dampeners

The pulsating gas delivery by a reciprocating compressor creates pressure variations in the suction and discharge lines. Suction and discharge "bottles" are used to smooth out these pressure surges. bottle furnish a .Quantity of gas that expands to retard pressure decreases and compresses to absorb pressure surges.

Lubrication

Proper lubrication is very important to good operation. Frame and cylinder/packing are lubricated by separate means. The extensive bearing surfaces must be provided with adequate lubrication to allow operation without excessive need for shutdown and maintenance. Lubrication is provided using an arrangement similar to that of an internal combustion engine. A sump below the compressor provides surge capacity for the oil pump that recirculates the oil through lines to small-diameter lubrication holes. Contamination of the compressed gas by lubricating oil can be a problem,
Nonlubricated units are available for unusual applications. Special packing and piston ring materials are required.

Sour Gases

When compressing sour gas:

(i) Follow NACE Standard MRO 175-88 when selecting materials. In particular, this means selecting a suitable high-grade stainless steel for all wetted surfaces. Also, avoid using any copper-based alloys.

(ii) Do not allow H2S to contaminate the crankcase oil. Gas leakage through the rod packing into the crankcase can be prevented either by using multiple

packing or by venting any leakage to the flare.

(iii) Send a gas analysis to the cylinder lubricating oil supplier and/or the compressor vendor. A special lube oil formulation may be required to provide adequate lubrication and to prevent washing of lube oil off the cylinder wall and scoring.

(iv) Use NACE certified instrumentation; install a corrosion coupon in each cooler to detect any corrosion.

Control Devices

Typical control applications include unloading for starting, regulating the compressor output to match the demand, and protecting compressor (and engine) from overload.

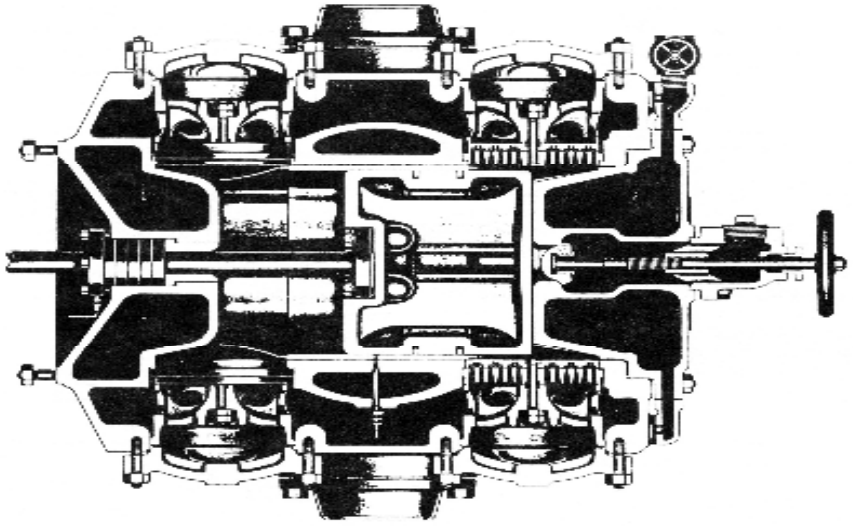


Figure 10-6a
Head-end fixed-volume clearance pocket (courtesy PETEX, 1987. Copyright © 1987 by the University of Texas at Austin).

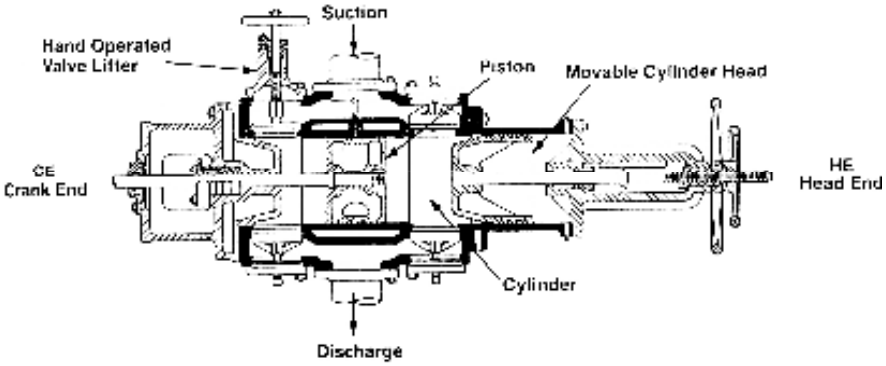


Figure 10-6b.
Cylinder equipped with a hand-operated valve lifter and variable-volume clearance (GPSA, 1987, p. 13-19).

Start-up unloading is mandatory for both manual and automatic starting to avoid exceeding available driver torque. Unloading is accomplished by discharge to suction bypass, discharge venting, or using valve lifters to hold inlet valves open.

Capacity control: At constant speed and compression ratio a reciprocating compressor admits the same volume of gas. Therefore, when the inlet pressure rises, more gas (weight, moles, scf) is compressed, and more brake horsepower (Bhp) is required. When appropriate, speed control for the prime mover is used. With steam or gas-engine drivers, the capacity is readily reduced by supplying less steam or fuel. When constant-speed electric motors are used as drivers, the capacity may be reduced by increasing the clearance as shown previously in Figure 10-7. Remote operation of clearance volume can be provided when controlled by valving, the operator must go to the compressor unit to adjust crank-driven clearance.

On-off control must be provided for prime movers. Starting a reciprocating gas engine requires a source of Compressed air or gas to initiate rotation of the drive shaft. As mentioned above, in compressor systems with several compressors in series and/or parallel, driver energy requirements should be minimized by proper selection of the compressor combination.

Compression Ratio

The compression ratio, r, of a single compressor stage (ratio of outlet to inlet pressure in absolute units) is limited to about 5:1 to 6:1 for two reasons: rod loadings and discharge gas temperature.

First, excessive rod loadings must be avoided to prevent mechanical failure. As shown in Figure 10-8, the rod load varies cyclically from compression to tension. The maximum rod loadings are (Fig. 10-8)

$$\text{Compression load} = P_d A_p - P_s (A_p A_r) \qquad (10\text{-}1)$$
$$= (P_d - P_s) A_p + P_s A_r$$

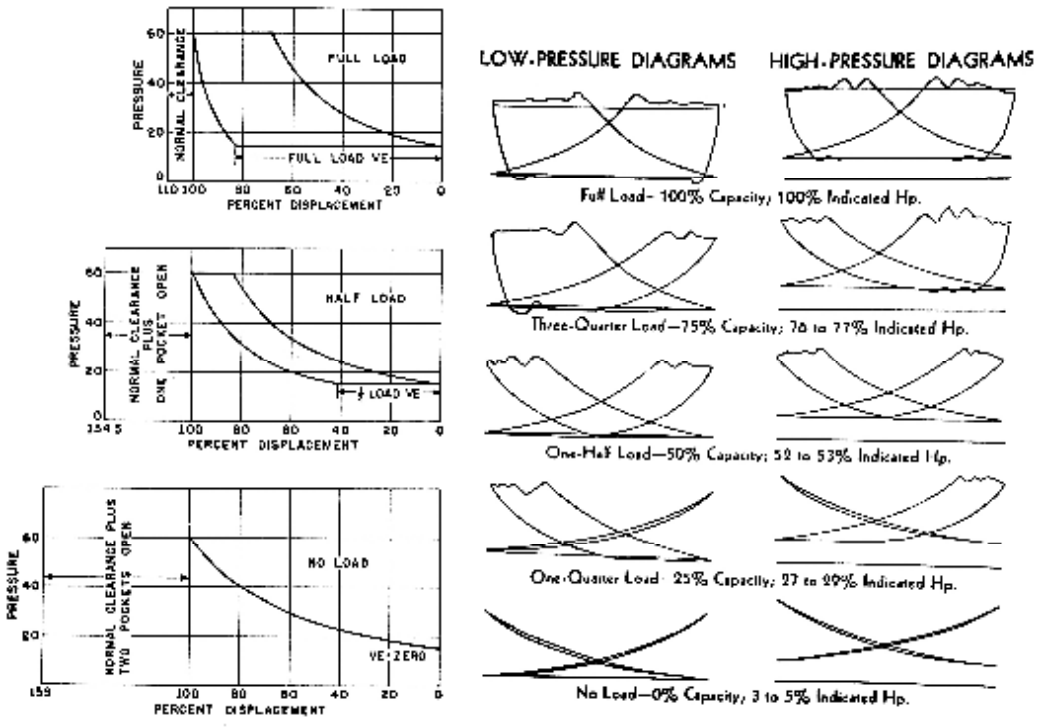

LOW-PRESSURE DIAGRAMS
HIGH-PRESSURE DIAGRAMS
PRESSURE
NORMAL CLEARANCE
FULL LOAD
FULL LOAD VE
PERCENT DISPLACEMENT
HALF LOAD
NORMAL CLEARANCE PLUS ONE POCKET OPEN
NO LOAD
NORMAL CLEARANCE PLUS TWO POCKETS OPEN
VE-ZERO
Full Load—100% Capacity; 100% Indicated Hp.
Three-Quarter Load—75% Capacity; 76 to 77% Indicated Hp.
One-Half Load—50% Capacity; 52 to 53% Indicated Hp.
One-Quarter Load—25% Capacity; 27 to 29% Indicated Hp.
No Load—0% Capacity; 3 to 5% Indicated Hp.

Figure 10-7
Reducing load by adding clearance (Ingersoll-Rand, 1980, pp. 4-6 and 4-7).

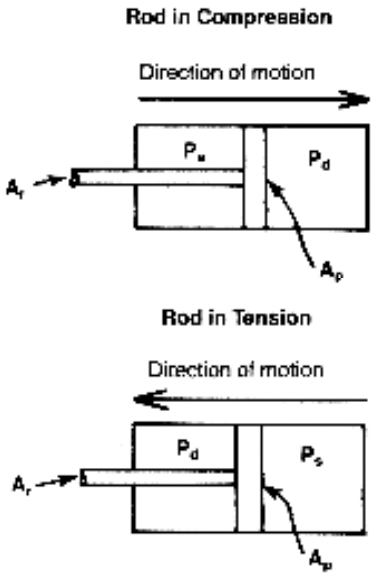


Figure 10-8.
Rod Loadings (GPSA, 1987)

$$\text{Tension load} = P_d (A_p - A_r) - P_s A_f \qquad (10\text{-}2)$$
$$= (P_d - P_s) A_p - P_d A_r$$

Where

P_d = discharge pressure
Ps = suction pressure
A_P = piston area
A_r = rod area.

Second, the gas is heated as it is compressed. The outlet temperature is normally limited to 300°F (150°C). Water or air cooling of the cylinders removes heat, but problems with lubricant breakdown and damage to compressor parts may still occur. If there is any doubt, consult the manufacturer.

High overall compression ratios are attained by using more than one compressor stage, or cylinder, with intermediate cooling of the process gas between stages and after the last stage. Figure 10-9 shows the P-V or indicator cards (or diagrams) for a two-stage compression. Figure 10-9 also shows how cooling (either by cylinder water jackets or by intercoolers) saves compression horsepower. Generally, intercooling savings are maximized by using equal compression ratios for all stages of compression. Figure 10-9 and the "equal r" rule-of-thumb both assume that no gas condensation occurs. When compressing natural gas, hydrocarbon liquids and possibly water can condense in the intercooler and must be completely removed in a scrubber before the next stage. Similarly, all liquids must be removed from the gas stream before it is fed to the first stage. It is very foolish to severely damage a million-dollar compressor as a result of attempting to save $1000 by undersizing the inlet separator or interstage scrubber.

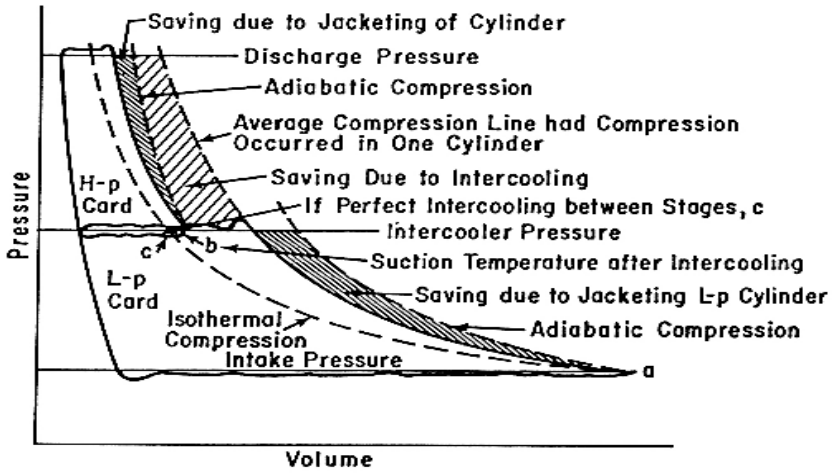


Figure 10-9.
Combined indicator cards from a two-stage compressor show how cylinder water jackets and intercooler help bring compression line nearer to isothermal (Miller, 1944).

Mounting and Housing

Reciprocating gas-engine/compressor units are large, noisy, and vibrating. Normal installation, especially for larger units, should be on a concrete foundation or pad. Some units are skid-mounted for portability and ease of interchange. Such an arrangement may be justifiable but will lead to increased maintenance, lubricant leakage, etc. Compressor units are normally housed inside buildings for cleanliness and protection against the environment. Also, the noisy exhaust piping of a gas-engine driver can be directed to the outside. Gas engines can be very noisy and ear protection is often required. Small field compressors are sometimes mounted under simple roofs with no walls.

Materials of Construction

Martino (1983) reviews the materials used in reciprocating compressors. Frames and running gear are not exposed to process gas and the materials used are fairly standard. The frame is gray cast iron. Crosshead extensions also are cast iron, either integral or bolted on. Crankshafts and connecting rods are forged steel. Bearings are sleeve type and generally aluminum.

Distance pieces separate the driver end and the compressor-cylinder (gas end) and provide a seal against entry of lube oil. These pieces are generally cast iron.
Gas-end materials are more of a problem because of the temperature and pressure ranges encountered, as well as possible corrosion problems with the process gas. In addition, there are a large number of parts including cylinders, cylinder liners and heads, pistons, piston rods, valves, rings, and packing.

Previously, cylinder parts were cast but, due to environmental problems with making castings, are now forged or otherwise fabricated. Cast iron can be used for most gases up to about 1500 psig, both lubricated and nonlubricated. Nodular iron is used at pressures up to 2200 psig and steel above 2200 psig. Cylinder liners are generally gray cast iron, except at low temperatures where Ni-resist is used.

Factors other than pressure may dictate cylinder materials. Stainless or 3.5% nickel steel are used below -60°F. Alloyed cast iron is used from -60 to -20°F. Generally, sour gas is handled with the aforementioned materials if the H_2S content is 5% or less. At higher concentrations, stainless steel is used.

Large pistons are made of aluminum to decrease inertia. Smaller pistons are made of cast iron or steel. Stainless steel is used for low temperature and sour gas service. Piston rods are alloyed carbon steel, with some exceptions.

Most packing cases are steel or cast iron, with stainless steel used for special applications. Piston rings are TFE (tetrafluoroethylene) for pressures to 5000 psig and bronze at higher pressures.

Valve material selection is rather complex due to severity of service and the need for springs. Different materials are used for the various parts. See Martino (1983) for further details.

ROTARY COMPRESSORS

Rotary compressors are also positive displacement machines. Primarily a special purpose compressor, they are not used as much as reciprocating. Nevertheless, rotary compressors are important and are described briefly for completeness. The basic types of rotary compressors include lobed blowers, screw-type, sliding vane, and liquid-ring compressors.

Lobed Blowers

A common type of lobed blower is the two-impeller type, having rotors or impellers with a figure-eight shape (Fig.10-10). the impellers rotate in opposite directions and entrap gas as shown in the figure, delivering it from suction to discharge. Contrary to the appearance of Figure 10-10, the rotors do not ride on each other or the compressor casing. One rotor is driven directly by the prime mover and the other is driven via a gear train to guarantee proper positioning. Small clearances are maintained between moving parts and the casing. Lobed blowers are inherently slow-speed, high-inertia devices. Typical applications are for rather large amounts of gases at low pressures and low compression ratios.

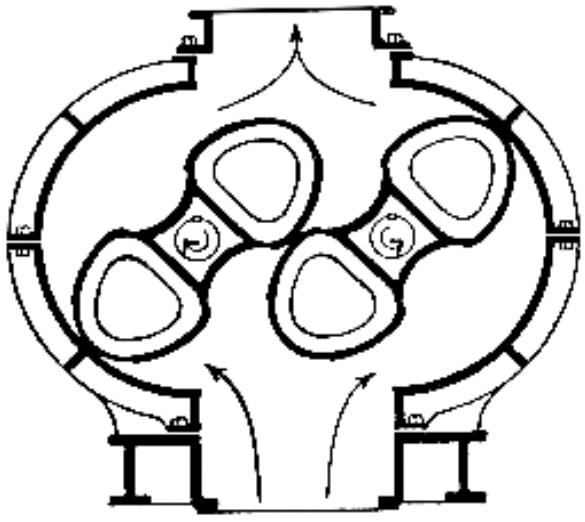


Figure 10-10
Two-impeller type of positive rotary blower (Perry and Green, 1984, p. 6-26).

Rotary Screw Compressors

Screw compressors are also called helical- or spiral-lobe compressors. Figure 10-11 depicts a typical unit. Their action is much like that of the lobed blower except that the helical design demands that the two rotors not be identical in shape. One rotor has rounded tips to fit into the rounded grooves of the other rotor. The rotors do not need to have the same number of lobes since they may operate at different rotation rates.

When the two grooves encounter their respective inlet ports, the opposite ends are closed off to their discharge ports. Each groove fills with inlet gas. As the two grooves rotate toward each other, the rounded lobe encounters the rounded groove and traps the inlet gas in the groove. Further rotation reduces the volume containing the gas, compressing it. When the discharge end of the groove reaches the outlet port, the compressed gas is delivered to the discharge pipe.

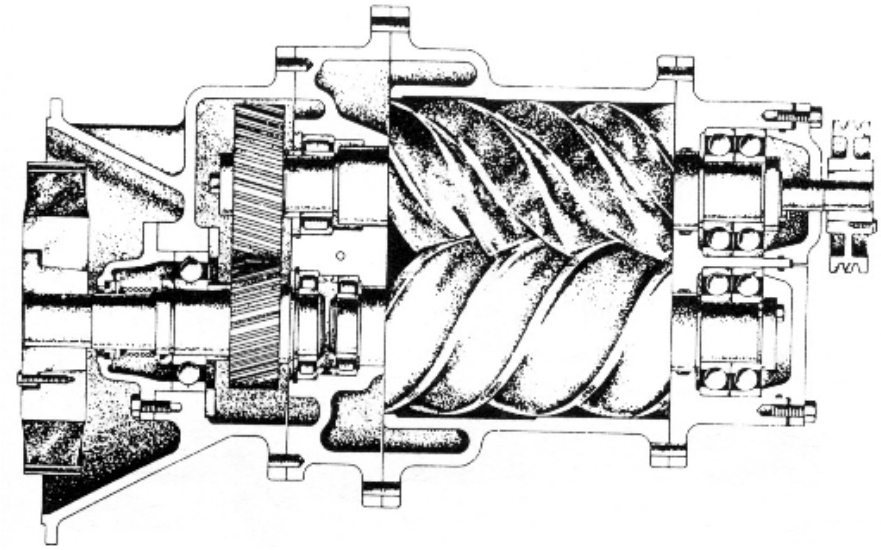


Figure 10-11
Cross section of typical single-stage oil-flooded screw compressor (Ingersoll-Rand, 1980, p. 8-2).

There is a parasitic back leakage of gas to suction that can be deliberately increased to reduce throughput. As with the lobed blower, one of the rotors is driven directly by the prime mover and the other is driven via a gear train. Screw compressors may be oil-flooded or dry. Oil in the former type lubricates the machine, furnishes a seal for the gas, and absorbs much of the heat generated during compression. This results in roughly "isothermal" compression and a lower Bhp. Oil-flooded rotary screws have 10 to 100% continuously-variable capacity control (by increasing the back leakage), easy start-up, and no pulsations. In air compression, where 2-3 ppm oil in the compressed air is acceptable for general shop requirements, rotary screws now dominate the market. Dry units have the advantage of delivering oil-free gas but

have particularly high noise levels unless inlet and outlet silencers are provided.

Both types are generally of low capacity, say a few hundred or thousand acfm. The units are higher in speed than the lobed blower and can attain higher compression ratios.

Sliding-Vane Compressor

The sliding-vane compressor consists of a rotor mounted eccentrically inside a casing. The rotor has longitudinal slots into which are fitted sliding vanes. Figure 10-12 is an end view of the compressor showing the action of the vanes in trapping the inlet gas and compressing it to the outlet port. In this respect, the sliding-vane compressor is like the reciprocating compressor, except that it needs no valves. Because of its design, however, the sliding-vane compressor has an internally fixed compression ratio. The exit gas, just before it goes to the exhaust port, has the same pressure regardless of the pressure in the discharge line, thus leading to inefficiency and limiting the pressure range.
Sliding-vane compressors are generally small units used for air compression or vacuum service. Speeds are moderate, roughly 600 to 1800 rpm. Drivers are usually electric motors directly coupled.

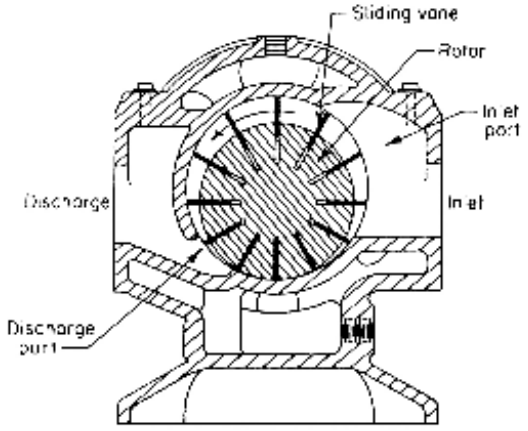


Figure 10-12.
Sliding-vane type of rotary compressor (Perry and Green, 1984, p. 6-26).

Liquid-Ring Compressors

Figure 10-13 shows a typical liquid-ring compressor. The rotor has forward-curved blades mounted on it that throw liquid against the outer casing, forming a liquid seal ring. The eccentric shape of the casing forces the liquid inward in the compression sector, compressing the gas and delivering it to the discharge line. Like the sliding-vane, the liquid-ring compressor has a fixed internal compression ratio. The liquid acts as lubricant, sealant, and coolant. Liquid-ring compressors are used for

corrosive gases, especially where oil cannot be tolerated. Other services include vacuum and vapor recovery.

CENTRIFUGAL COMPRESSORS

Dynamic compressors consist essentially of one or more impellers mounted on a shaft that rotates at 2000 to 20000 rpm within a casing (Fig. 10-14). Stationary diffusing passages and guide vanes connect the flow between impellers. Each compressor "stage" includes one group of rotating impeller and stationary diffuser and guide vanes. The two common types of dynamic compressors-centrifugal and axial-accelerate the fluid in the radial and the axial directions, respectively. Some blade designs place a compressor in an intermediate category called "mixed-flow."

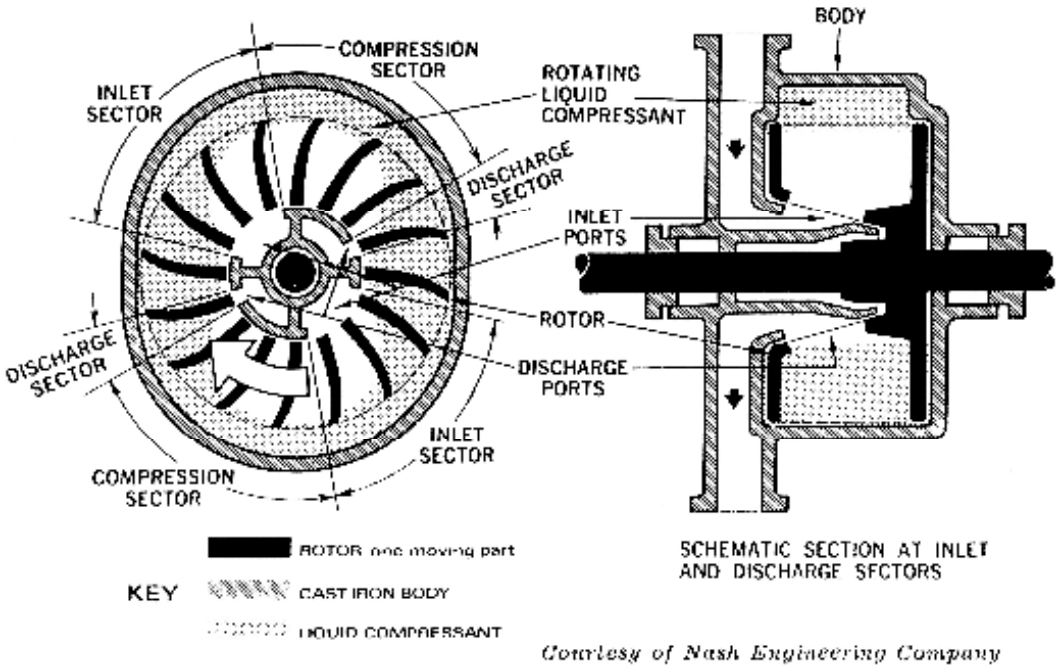


Figure 10-13
A typical liquid-piston rotary compressor (Copyright © 1970, The Nash Engineering Company).

When properly balanced and lubricated, centrifugal compressors exhibit little wear and tear. Typical operating time between shutdowns for routine maintenance is 1-2 years. The units are relatively lightweight and essentially vibration-free. Cheaper mounting is possible than for reciprocating compressors.

Mechanical Details

Figure 10-14 shows a typical multistage centrifugal compressor. Several impeller wheels are mounted on a rotating shaft supported horizontally on lubricated

bearings. Casing-shaft seals are needed to prevent leakage of the process gas outwards. Since the gas flow is continuous, there is no need for inlet and outlet valves. In some applications, adjustable inlet guide vanes are used to control throughput.

Multiple wheels are often used since the compression ratio of a single wheel is limited. Up to 10 impellers may be used in a single frame. Some of these may be in series (to increase the compression ratio) and some in parallel (to increase capacity). Two or three frames in series may be required for high-pressure applications. Intercooling of a hot gas stream can be done for a side stream from a single frame (i.e., between wheels) or between frames.

The impeller of the centrifugal compressor operates very much like that of a centrifugal pump. Gas enters in the eye of the impeller and is accelerated outward by the motion of the impeller blades. A portion of the imparted kinetic energy is transformed into pressure as the gas expands in the impeller. An additional smaller pressure rise occurs in the diffuser following the impeller.

A limitation in the operation and design of centrifugal compressors is imposed by the phenomenon known as "surge." Surge is an aerodynamic condition occurring at low flow and constant speed in which vibration may cause mechanical damage to the compressor. Obviously, it is absolutely imperative to prevent surge as is discussed under controls.

Impellers

The rotating impellers impart kinetic energy to the gas. Thus, impeller design is of great importance (Sayyed, 1976).

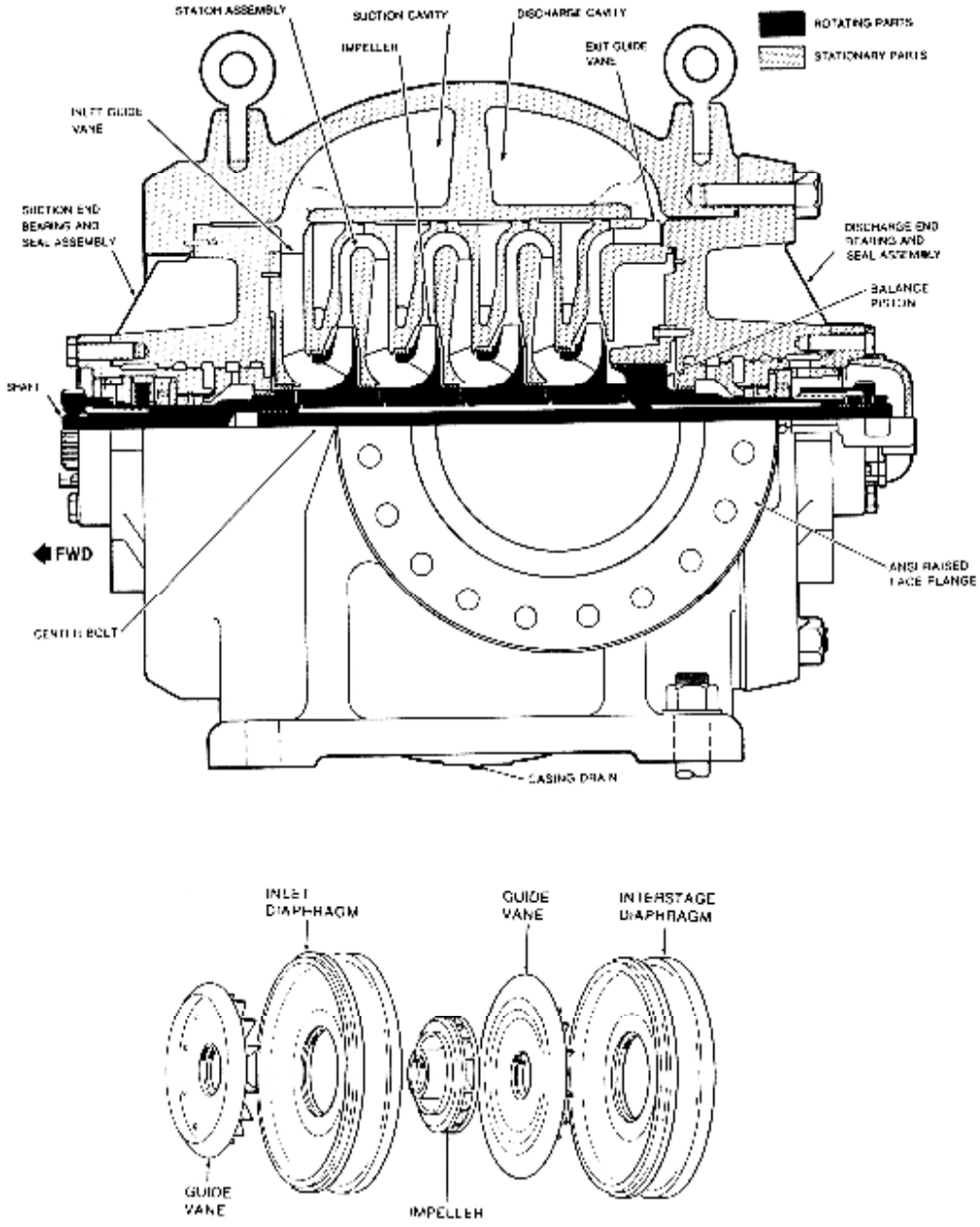


Figure 10-14
Solar centrifugal compressor (Solar Turbines, Inc., 1989).

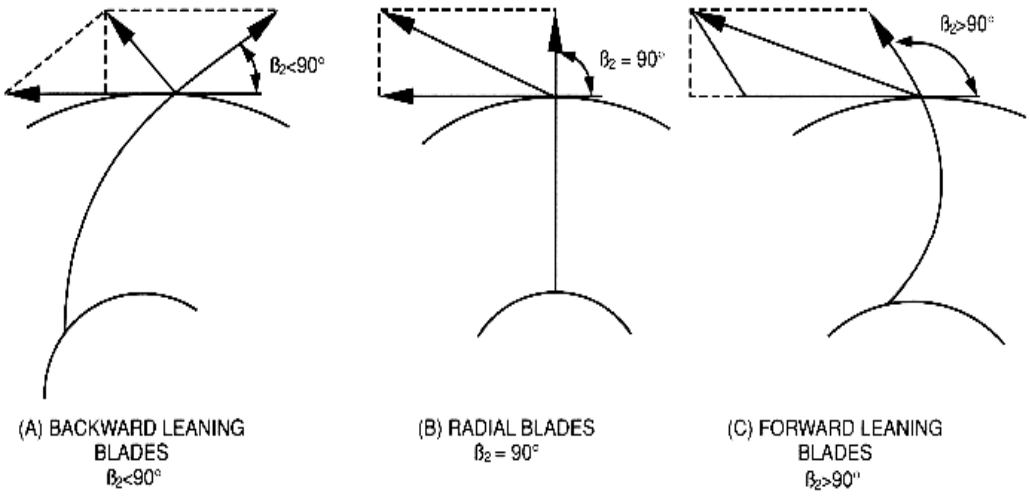


Figure 10-15
Centrifugal compressor impeller blade configurations (Sayyed, 1976).

Impeller blades usually lean backward (Fig. 10-15). The blades may be open or shrouded (Fig. 10-16). Open impellers have excellent operating characteristics but require closer tolerance with the stationary parts than shrouded impellers. Shrouded or closed impellers are made in three basic styles: three-dimensional, two-dimensional, and wedge (Fig. 10-17). Three-dimensional impellers have wider flow ranges than two-dimensional but are more expensive. Two-dimensional impellers are used mainly for low flow rates while wedge blades are used for very low flows.

Stationary Parts

Nozzles, diaphragms, diffusers, and guide vanes are included in the stationary parts. These parts guide the gas into the impellers, channel it between impellers, and deliver it to the discharge chamber.

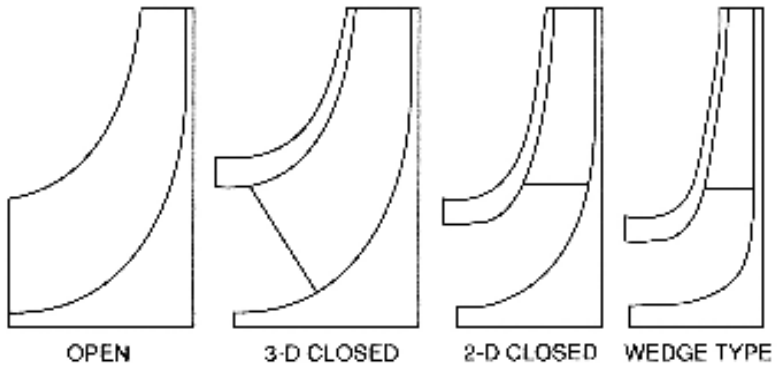


Figure 10-16
Centrifugal compressor impeller types (1976).

Inlet nozzles accelerate the fluid into the impeller with minimum friction loss and provide a uniform velocity profile for the gas entering the impeller. Inlet nozzles may be axial or volute. Axial nozzles provide better velocity profiles, especially at high specific speeds, but require more space than volute nozzles (Sayyed, 1976).

Diffusers are expanding flow channels that guide the gas from the exit of one impeller to the inlet of the next and convert some of the kinetic energy imparted by the impeller into pressure energy. Diffusers may have vanes for improvement of aerodynamic behavior, but vanes limit the flow range greatly and are often omitted. Outlet nozzles are special diffusers that convey the gas to the discharge cavity and are axial or volute.

Diaphragms are the solid pieces that form the walls of the channels constituting the nozzles and diffusers.

Guide vanes are sometimes used in the inlet nozzle to provide a desired flow direction to the impeller inlet. Guide vanes may be fixed or adjustable. The latter are sometimes used to provide a measure of flow control through the compressor.

Casings

The centrifugal compressor casing houses the stationary and rotating parts and forms the outside wall of the inlet and outlet chambers. Two basic types of casing are used, the horizontally split casing and the vertically split or barrel casing.

Figure 10-18 depicts a typical horizontally split casing with the top half removed. The two pieces of the casing are bolted together. This is the preferred type of casing for single or multistage compressors. The chief limitation on the horizontally split casing is its low pressure rating for higher pressures, the barrel-type casing is superior (Fig. 10-19a). The barrel casing consists essentially of a cylindrical shell with end covers bolted at low pressures (Fig. 10-19b) and held in place by shear rings at high pressures (Fig. 10-19c).

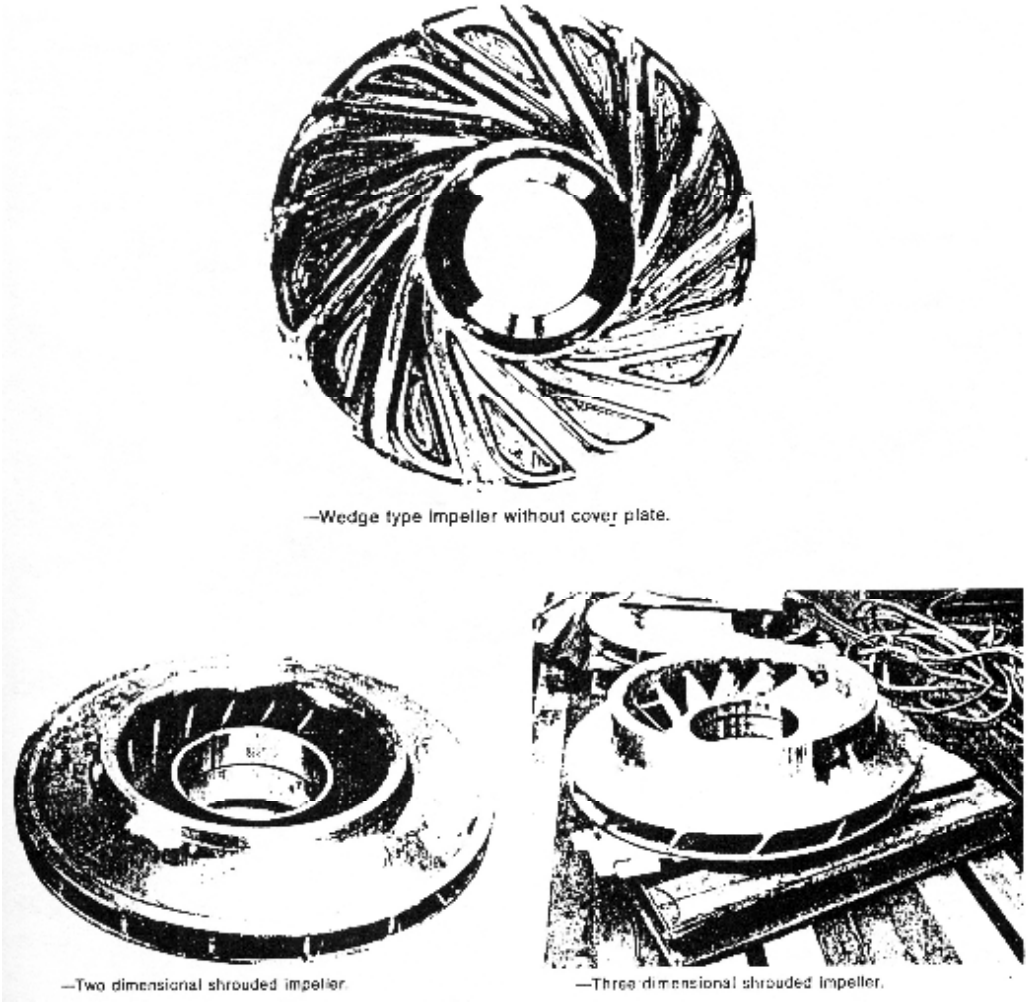


Figure 10-17
Centrifugal compressor impeller designs (Sayyed, 1976).

Bearings

The main hearings on a centrifugal compressor rotor provide a lubricated suspension for the rotor. These bearings are generally located outside the casing to prevent oil contamination of the process gas and vice versa. The bearings are of the sleeve, ball, or roller bearing types, depending on the service. A thrust bearing is provided on the low-pressure end of the rotor to absorb unbalanced pressure force in the axial direction.

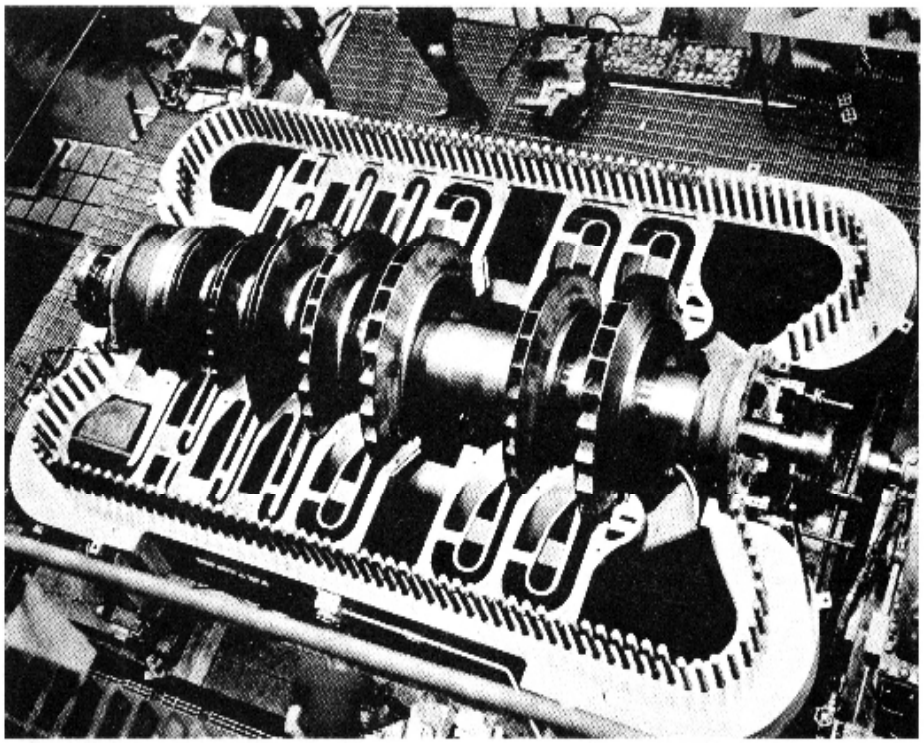


Figure 10-18.
Horizontally-split centrifugal compressor (Neerken, 1975, Vol. 95, No. 10, 22-26, 1988).

The pressures on either side of each impeller wheel are not much different. The axial thrust on the impeller is caused mostly by the larger area on the back side of the impeller than on the front side. A so-called balancing drum is placed on the discharge side of the rotor, with the high pressure on the inside and suction pressure on the outside (Fig. 10-20). This approximately balances the pressure forces exerted on the impellers. An axial thrust bearing is provided on the low-pressure end of the rotor to absorb the remaining force.

Seals

The pressure differences between the inlet and outlet sides of each impeller and between the inside and outside of the casing provide strong driving forces for leakage, thus requiring seals such as those shown in Figure 10-21. The labyrinth seal (Fig. 10-20) is invariably used for interstage

Applications. The arrangement in Figure 10-21 has the advantage that the casing-to-atmosphere seal must withstand only the suction pressure at both ends of the compressor. Because labyrinth seals always leak process or seal gas, they are not used as casing-atmosphere seals except for air whose leakage to atmosphere poses no problems. Mechanical (contact) seals operate with zero clearance and oil pressures 35-50 psi

above the internal gas pressure. Liquid film seals (pumping and cylindrical bushings) are suitable for severe service but require higher oil circulation than the mechanical contact type. The sleeves must be lined with Babbitt or other non-galling material that is compatible with both the gas being compressed and the circulating oil. The circulating oil acts as the sealing fluid, cools the seal area, and lubricates the sleeves.

Balancing

Centrifugal compressors are high-speed devices, rotating in the range of 5000-20000 rpm. Rotating shafts have critical speeds corresponding to their natural vibration frequencies. Lateral vibration at these srjgeds will be multiplied by resonance with the shaft, and structural failure can result. Normal rotor speeds exceed the first or higher critical speeds, thus demanding that a rotor be properly balanced when installed to minimize vibration.

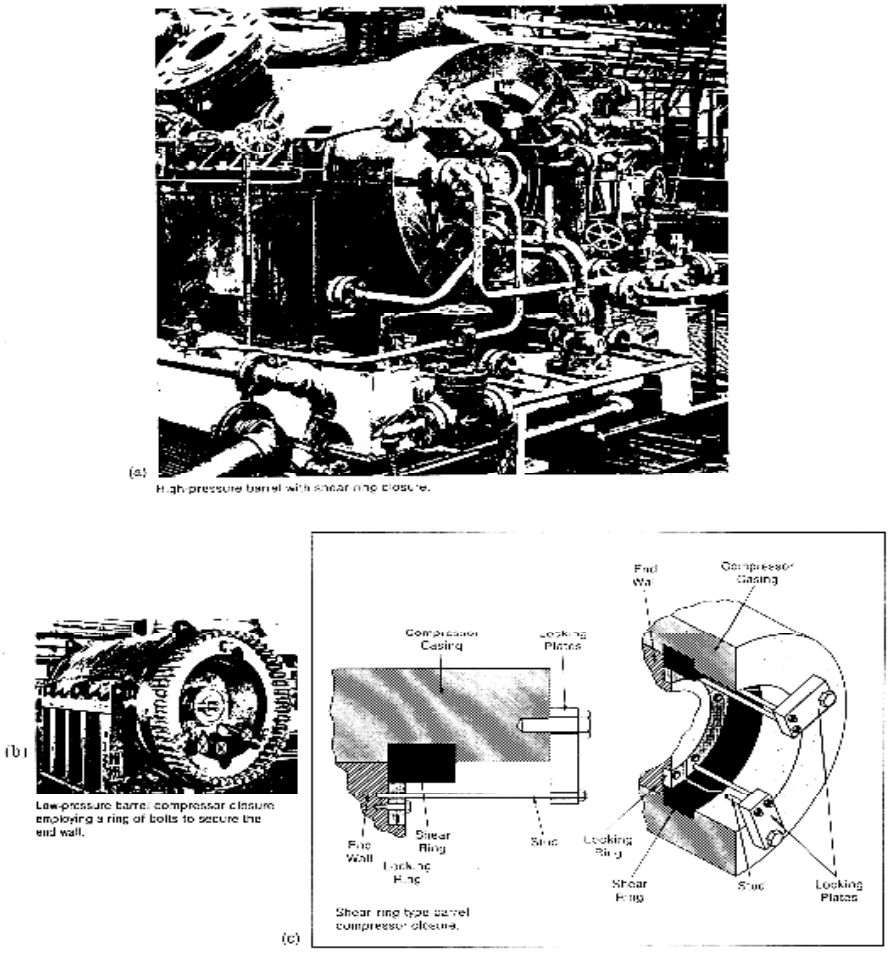


Figure 10-19
Barrel-type casing (Elliott Co., 1981).

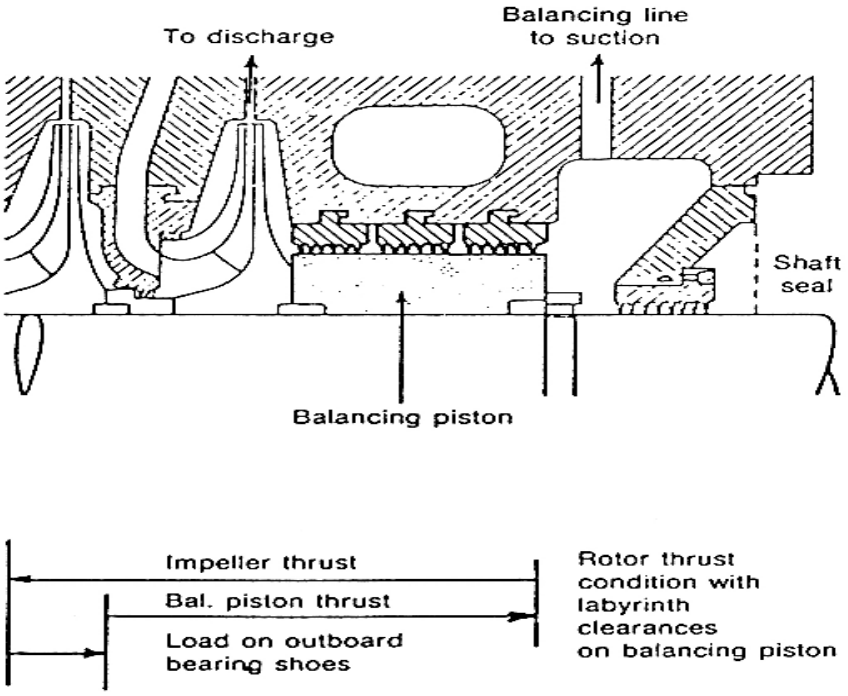


Figure 10-20
Centrifugal compressor balancing piston (Nelson, 1977).

Compression Ratio

The compression ratio of a single impeller is inherently limited to 1.2-1.5. However, the overall compression ratio of a compressor can be increased by adding impellers in series.

The temperature limit is normally about 300°F, as dictated by compressor materials. Higher temperatures can be tolerated with proper selection of materials.

Materials

Selecting materials to use in turbomachinery (dynamic compressors and turbines) is very complex. Many factors must be considered such as operating stresses of rotating and stationary components, corrosion, and operating temperatures (Cameron et al. 1982).Appropriate construction materials are listed in API Standard 617, Appendix B.

Centrifugal compressor impellers are usually made from alloy steel such as AISI 4140 or 4340. Blades can be fillet welded to the hub and cover (fully closed impeller). Impellers are heat treated to provide the desired hardness and strength. Consult Cameron et al. (1982) for further details.

Controls

Figure 10-22 presents typical characteristic curves for a centrifugal compressor and shows how the gas flow rate, head generated (discharge pressure-suction pressure) and speed are related. At constant speed, a compressor will not operate stably below a certain minimum flow called the surge point. Surge occurs when the compressor cannot generate the head required overcome the imposed discharge/suction pressure ratio. As the flow decreases, partial stall (or minor recirculation near the impeller tip) occurs first. If the gas throughput is decreased further, violent surge or oscillation of the entire flow in the compressor occurs. Surge is very harmful to the compressor (Sapiro) because:

1. Rotor vibration can damage interstage labyrinth seals.
2. Flow reversal continuously increases temperatures in the compressor by bringing higher temperature gas into the compressor. Such overheating can damage sealsjmd bearings.
3. Rapid changes in axial thrust may damage thrust bearings.

Two types of controls are needed: process and anti-surge. Process controls regulate the throughput, discharge pressure, and suction pressure in combinations consistent with the given compressor performance characteristics as shown in Figure 10-22. At a given compressor speed, setting any two of the three process variables fixes the third. Antisurge control ensures the flow always exceeds the surge limit.

There are four basic control jnethods:

1. Suction throttling
2. Discharge throttling
3. Moveable inlet guide vanes in compressor
4. Driver speed regulation.

For minimum energy consumption, driver speed regulation is best. Variable-speed drivers are not always practical. Overall economics may favor constant-speed drivers, in which case the other three methods are used. • Anti-surge minimum flow limitcr control continously measures compressor flow and head to detect proximity to the surge line, i.e., minimum flow line. At a preset flow—surge minimum plus a safety allowance—the control opens a bypass valve downstream of the discharge cooler to recycle gas. By cooling the recycle gas, it is possible to run with 100% recycle without overheating the compressor.

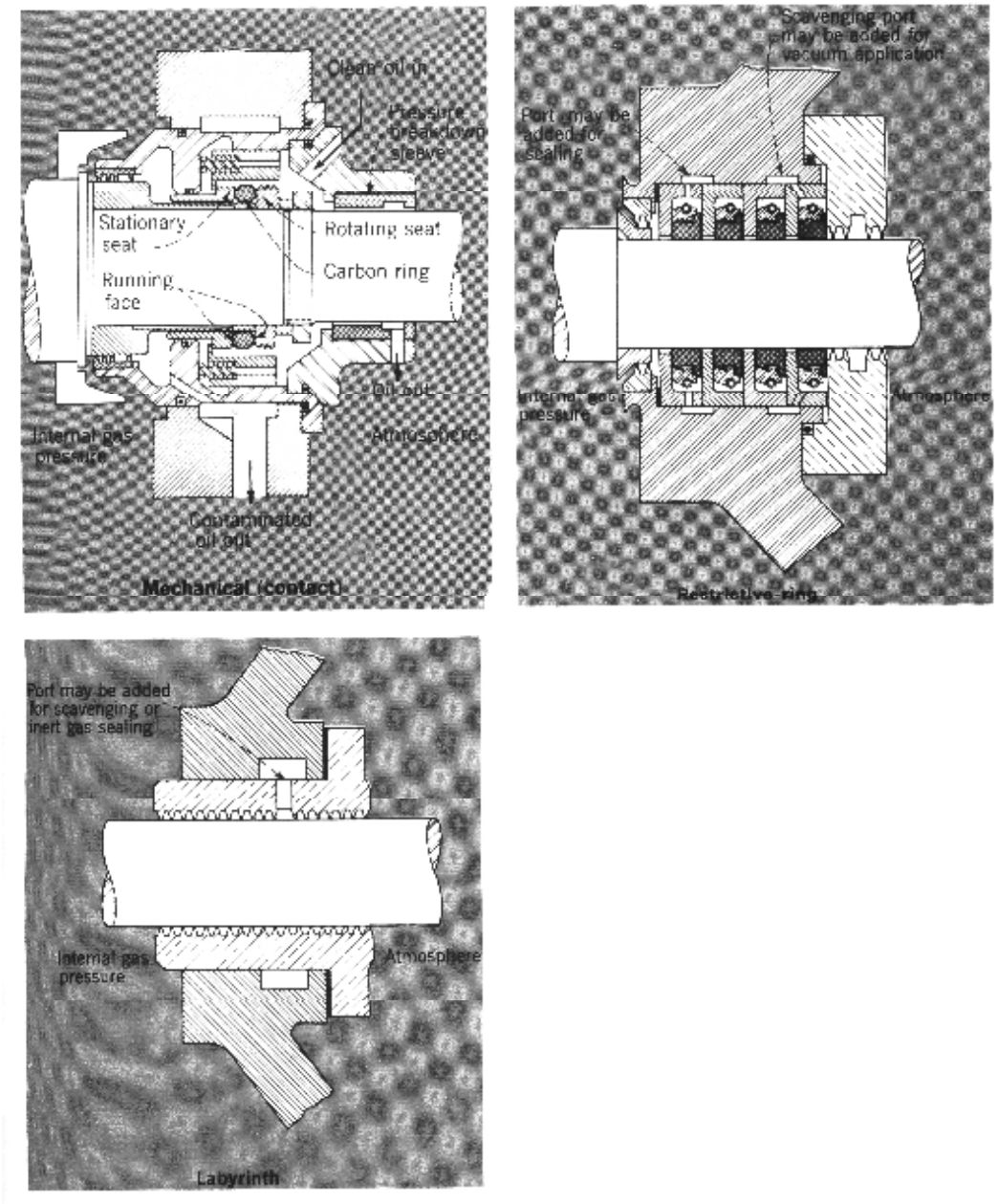


Figure 10-21
Shaft end-seals for centrifugal compressors (API std617).

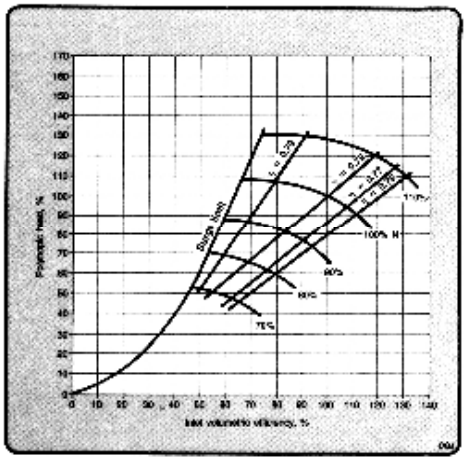


Figure 10-22
Typical centrifugal compressor head-capacity curves (Campos & Rodrigues, 1988).

Testing

Larger centrifugal compressors should pass a performance test in the manufacturer's shop before shipment. Performance shortfall experience suggests that the added expense is justified. The performance test should follow ASME Power Test Code 10 and be witnessed (Hunt, 1989).

COMPRESSOR DESIGN

The basic design equation is presented first, and the isotropic and polytropic compression models are described next. Then these models are applied to reciprocating and centrifugal compressors and some design examples given. Finally, volumetric efficiencies and water requirements of reciprocating compressors are discussed.

Basic Relations

Compressor design is based on the first and second laws of thermodynamics. The "first law" or energy balance is

Written for the steady-state gas flow through the compressor. If kinetic and potential energy changes are neglected, as well as heat loss to the surroundings, then:

$$-W = h_2 - h_1 \qquad (10\text{-}3)$$

Where

- W = compressor work, Btu/lbm or kJ/kg (negative because the work is done on the gas)
- h_2 = discharge gas enthalpy, Btu/lbm or kJ/kg
- h_1 = inlet gas enthalpy, Btu/lbm or kJ/kg

Use of Equation 10-3 requires that the inlet pressure and temperature and the outlet pressure and temperature be known to evaluate the change in enthalpy. However, in design work only the first three of these items are known. The outlet temperature cannot be arbitrarily specified. A compression model is needed.

Isentropic Model

This classic model assumes "second-law perfection"—the gas is compressed reversibly and adiabatically; that is, at constant entropy (isentropically).

$$-W_{is} = h_{2s} - h_1 \qquad (10\text{-}4)$$

Where

- $-W_{is}$ = isentropic work, Btu/lbm or kJ/kg
- h_{2s} = discharge gas enthalpy for a constant-entropy path, Btu/lbm or kJ/kg

An experimentally-determined compression efficiency, ir)is, is now used to relate the theoretical isentropic work, W_{is}, to the actual work, W_{act}, required by the compressor

$$-W_{act} = -W_{is}/\eta_{is} \qquad (10\text{-}5)$$

Where

- $-W_{act}$ = actual compression work, Btu/lbm or
- kJ/kg η_{is} = isentropic efficiency, dimensionless

The compressor "gas" power is obtained by multiplying the work per unit mass by the mass flow rate.

$$p_g = m(-W_{act}) \qquad (10\text{-}6)$$

Where

- p_g = gas power, Btu/hr or kJ/s (kW)
- m = mass flow rate, Ibm/hr or kg/s

$$\text{Gas horsepower, Ghp} = p_g\,(\text{Btu/hr})/2544 \qquad (10\text{-}7)$$
$$= p_g\,(\text{kW})/\,(.746)$$

The isentropic efficiency accounts for irreversibilities that occur as the gas is being compressed and are probably dissipated as turbulent friction within the gas. In addition, mechanical inefficiency occurs in the compressor due to rolling friction of the shaft in its bearings and the like. These inefficiencies are handled later. The total shaft power required by the compressor from the prime mover is called the brake power or Bhp.

The above equations may be applied directly to substances for which thermodynamic data are available in charts, tables, or computer programs.

For substances for which complete P-T-h-s data are not easily available, but for which ideal gas behavior is approached, simple equations can be derived. These equations are suitable for hand calculations. Many permanent gases approach ideal gas behavior at ambient and higher temperatures if the pressure is not too high. For an ideal gas:

$$-W_{is} = h_{2s} - h_1 = C_p(T_{2s}-T_1) \qquad (10\text{-}8)$$

Where

C_p = mean heat capacity at constant pressure between T_1 and T_{2s}, Btu/lbm-R or kJ/kg-K

T_1 = inlet gas temperature, °R or K

T_{2s} = discharge gas temperature, isentropic path, °R orK

The isentropic coefficient, k, is defined by

$$K = C_p / C_v \qquad (10\text{-}9)$$

Where C_v = constant volume specific heat, Btu/lbm-R or kJ/kg-K

For an ideal gas,

$$C_v = C_p - R/MW \qquad (10\text{-}10)$$

Where R = gas-law constant, energy/mol degree

MW = molecular weight of the gas

Combining Equations 10-9 and 10-10 gives:

$$Cp = R\,k / [MW\,(k-1)] \qquad (10\text{-}11)$$

Finally the isentropic discharge temperature for an ideal gas is:

$$T_{2s} = T_1\,(P_2/P_1)^{(k\sim1)/k} \qquad (10\text{-}12)$$

Where P_2 = discharge pressure, psia or kPa

P_1 = inlet or suction pressure, psia or kPa

Substituting Equations 10-11 and 10-12 into Equation 10-8 yields:

$$-W_{is} = \frac{RkT_1}{MW(K-1)}[(P_2/P_1)^{(K-1)/K} - 1] \qquad (10\text{-}13)$$

To calculate $-W_{is}$ in Btu/lbm, use R = 1.986 Btu/lbmol °R, T in °R, and P in psia. For -Wis in kJ/kg, use R = 8.3145 kJ/kgmol K, T in K, and P in kPa.

Non-ideal gas behavior is allowed for empirically by:

$$-W_{is} = Z_{av}\frac{RKT_1}{MW(K-1)}[(P_2/P_1)^{(K-1)/K} - 1] \qquad (10\text{-}14)$$

Where $Z_{av} = (Z_1 + Z_2)/2$ (10-15)

= average compressibility for the gas

Equation 10-14 is used henceforth exclusively. If the gas is approximately ideal, simply set $Z_{av} = 1$.

A commonly encountered form of Equation 10-14 involves substituting 1545 ft Ibf/lbmol °R for the gas-law constant R. Thus, the work is in ft Ibf/lbm, the so-called adiabatic head. The term "head" or "ft" is used because 1 Ib mass of gas weighs 1 Ib wt and so ft Ibf/lbm can be considered ft Ibf/lb wt or "ft."

$$H_{is} = Z_{av}\frac{1545KT_1}{M(K-1)}[P_2/P_1)^{(K-1)/K-1]} \qquad (10\text{-}16)$$

Where H_{is} = adiabatic head, ft Ibf/lbm or "ft"

The gas horsepower, Ghp, is found by using the appropriate conversion factor.

$$Ghp = rhH_{is}/(l{,}980{,}000\ \eta_{is}) \qquad (10\text{-}17)$$

Equation 10-12 does not estimate the actual discharge temperature of a compressor. A safe estimate assumes that all the irreversibilities in the compressor increase the gas enthalpy and thus the temperature. The gas discharge temperature is:

$$T_2 = T_1 + T_1\,[(P_2/P_1)^{(K-1)/K} - 1]/\ \eta_{is} \qquad (10\text{-}18)$$

In practice, Equation 10-18 is often used with Equation 10-14, whether or not $Z_{av} = 1$. Note that both isentropic and actual compressions are assumed to be adiabatic. Therefore, if the mean heat capacity, C_p, is the same over the two temperatures ranges-T_1 to T_{2is} and $T_1$to T_2 -then:

$$\eta_{is} = (T_{2s} - T_1)/(T_2 - T_1) \qquad (10\text{-}19)$$

Some authors (Cohen et al., 1975) use Equation 10-19 rather than Equation 10-5 to define η_{is}.

Polytropic Model

A second compression model uses a polytropic coefficient, n, in place of the isentropic coefficient, k. By definition, n provides the actual outlet or discharge gas temperature, T_2.

$$T_2 = T_1 (P_2/P_1)^{(n-1)/n} \qquad (10\text{-}20)$$

However, compressor manufacturers do not supply values for n; instead they provide a polytrophic (or small-stage) efficiency, η_p By definition, TIP is the isentropic efficiency of an elemental stage in the compression such that η_p is constant throughout entire compression (e.g., a multistage centrifugal compressor). Mathematically

$$\eta_p = dT_s/dT \qquad (10\text{-}21)$$

dT_s = elemental temp rise for isentropic compression, dP
dT = elemental temp rise for actual compression, dP

Integration of Equation 10-21 and use of Equation 10-12 yields:

$$T_2/T_1 = (P2/P1)^{(k-1)/k\eta_p} \qquad (10\text{-}22)$$

Comparison of Equations 10-20 and 10-22 shows:

$$(n\text{-}l)/n = (k\text{-}l)/(k - \eta_p) \qquad (10\text{-}23)$$

The adiabatic polytropic work or head is

$$-W_p = Z_{av} \frac{RnT_1}{MW(n-1)}[(p_2/P_1)^{(n-1)/n} - 1] \qquad (10\text{-}24)$$

Then the actual work of compression, Wact, is:

$$-W_{act} = -W_p/\eta_p \qquad (10\text{-}25)$$

Finally, the isentropic and polytropic efficiencies are related by:

$$\eta_{is} = \eta_p[(p_2/p_1)^{(K-1)/K} - 1]/(p_2/p_1)^{(n-1)/n} - 1] \qquad (10\text{-}26)$$

Sizing (Estimating BHP)

Reciprocating compressors are usually sized using the adiabatic model and adiabatic efficiency. The mechanical efficiency is about 0.93. In other words, the brake power is given by the gas power divided by 0.93. Isentropic efficiencies for reciprocating compressors run fairly high, say 0.88-0.90. Contact a reliable manufacturer for more accurate sizing.

Reciprocating compressor horsepower can be estimated by three methods in addition to those previously described (GPS A, 1987).

1. Quick Estimate (GPSA, 1987, p. 13-4, Eq. 13-4)

$$BHP = (MF)(r)(N)(MMcfd)(F) \quad (10\text{-}27)$$

Where BHP = brake horsepower
MF = multiplication factor
= 22 for large, slow speed (300-450 rpm)

Units with SG = .65 and r > 2.5
= 20 when 0.8 < SG < 1.0
= 16 to 18 when 1.5 < r < 2
r = compression ratio per stage
N = number of stages
MMcfd = capacity at 14.4 psia and Ts
F = 1.0 for single-stage
= 1.08 for two-stage
= 1.10 for three-stage
T_s = suction temperature (°R).

2. Quick Graphical (see GPSA, 1987; p. 13-7, Fig. 13-9).

3. Detailed Estimate (GPSA, 1987; pp. 13-11 through 13-14)
   a. Calculate r for each stage.
   b. Estimate average $k = C_p / C_v$ for each stage using GPSA, 1987, Fig.13-8, p.13-6.
   c. Estimate discharge temperature, Td, for each stage using Equation 10-12 or GPSA, 1987 Figure 13-11 or Equation 13-18.

$$T_d = T_s + T_s [r^{(k-1)/k} - 1] \quad (10\text{-}28)$$

   d. Estimate compressibility factors at suction, Z_1 and discharge, Z_2, conditions and then calculate average $Z_{av} = (Z_1 + Z_2)/2$.
   e. Calculate Bhp using GPSA, 1987, Equation 13-21, p. 13-11.

$$Bhp = (Bhp/MMcfd)(P_L/14.4)(T_s/T_L) \times (Z_{av}) (MMcfd) \quad (10\text{-}29)$$

Where Bhp/MMcfd is read from Figures 13-12, 13-13
P_L = standard pressure, psia, for MMcfd
T_L = standard temp, °R, for MMcfd

f. Corrected estimate

$$BHP (corr) = Bhp (Eq. 10\text{-}29) (CHS)X (CLIP) (CG) \quad (10\text{-}30)$$

Where CHS = correction for high speed, p. 13-11
CLIP = correction for low intake pressure, GPSA, 1987, Figure 13-14, p.13-14
CG = correction for gas gravity,

GPSA, 1987, Figure 13-15, p. 13-14.

Centrifugal Compressors

Both the adiabatic and polytropic models can be used; however, the polytropic efficiency varies less with compression ratio. Centrifugal compressor efficiencies are lower than those for recips, of the order of 0.75. The Elliott table and chart (Fig. 10-23) can be used for estimation purposes. First find the polytropic efficiency in the table, then use the chart to find the adiabatic efficiency. Mechanical efficiencies are not generally used to determine brake power for centrifugal compressors. A rough rule is to simply add 50 HP for bearing and seal losses.

Pressure-Enthalpy Diagram. This method is the easiest to apply, involving straightforward use of the log P-h diagram. However, log P-h diagrams are generally available only for pure components, which limits the utility of the method. The method is now illustrated.

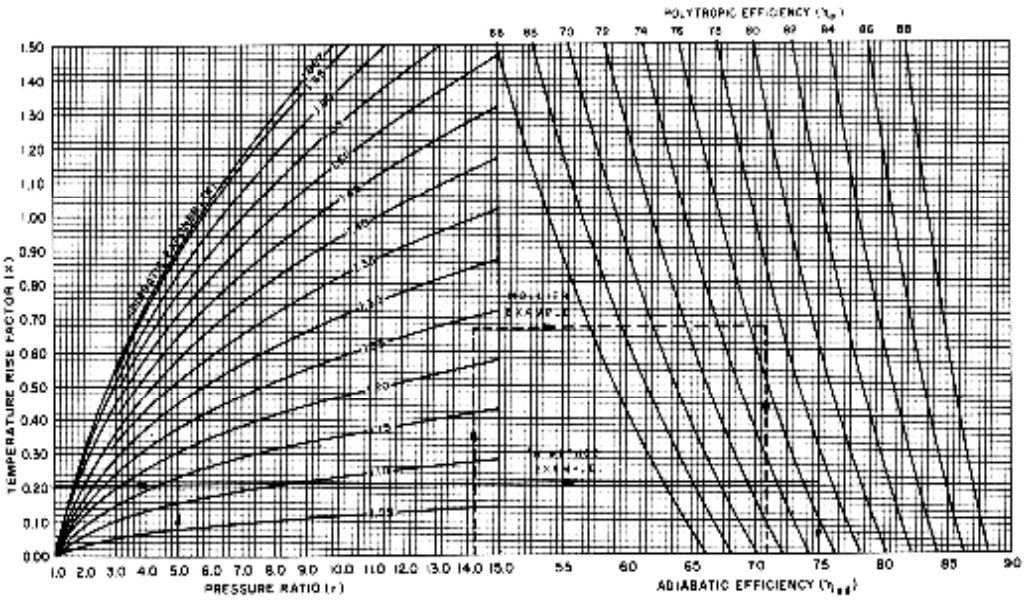


Figure 10-23
Elliott compressor data (Elliott Co., 1981).

Example 10-1. Compress 50,000 Ibm/hr propane from 20 psia, 80°F, to 100 psia in a centrifugal compressor. The adiabatic efficiency is estimated to be 0.75.

Solution: See Figure 10-24 for the log P-h diagram.Locate the point 20 psia, 80 F, marked 1, for which h, = -651 Btu/lbm Now follow a line of constant s upward and to the right to 100 psia, as indicated by point 2s in Figure 10-24.

H_2S = -611 Btu/lbm
$-W_s = h_{2s} - h_1$
= -611 - (-651) = 40 Btu/lbm
$-W_{act}$ = 40/0.75
= 53.3 Btu/lbm
$h_2 = h_1 + (-W_{act})$
= -651 + 53.3 = -598 Btu/lb

This point is located on Figure 10-24 as point 2.
From the graph T_2 = 211°F.

The power is now computed.
P_c = 50,000 x 53.5/2544 + 50 = 1100hp <—
Log P-h charts can be used for gas mixtures that are essentially pure—say 95 mol Percent or better. When feasible, this method is the easiest, and most accurate.

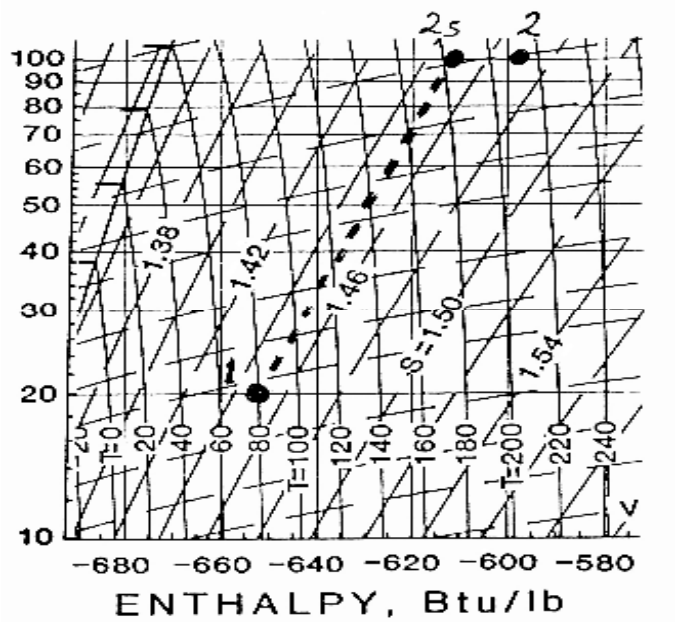


Figure 10-24
Compressor design using log P-h diagram (GPSA, 1987, p.24-30, Fig. 24-6, drawn by B. C. Bain and J. F. Ely, NBS, Boulder, CO).

Thermodynamic Tables

Starling (1973) presents tables of thermodynamic properties for a number of hydrocarbons. A portion of the propane table is shown below.

Psia		80°F	Psia		180°F	200°F	220° F
15	V	8.60678	100	V	1.45359	1.51021	1.56590
	h	-652.20		h			
					-614.16	-604.29	-594.23
	S	1.4608		S	1.4439	1.4592	1.4744
25	f/p	0.9833		f/p	0.9372	0.9434	0.9488
	h	-653.16					
	S	1.4365					
	f/p	0.9720					

Example 10-2. Solve the previous problem using the tabular data of Starling.

Solution: Find h_1 and S_1 by interpolation of the first column of data at 80F
Using linear interpolation for h_1:
h_1 = (-652.20-653.16)/2
= -652.68 Btu/lbm

For an ideal gas s + R In P is constant at constant temperature—therefore linear interpolation of s versus P at constant temperature is inaccurate.

s (15 psia) = 1.4608 Btu/lbm °R
(R/MW) In 15 = (1.987/44) In 15
= 0.1220 Btu/lbm °R
s + (R/MW)ln 15 = 1.5828 Btu/lbm °R Similarly
s + (R/MW)ln 25 = 1.5816 Btu/lbm °R Linear interpolation yields:
s + (R/MW) In20 = 1.5822 Btu/lbm °R and finally
s (20 psia, SOT) = 1.4472 Btu/lbm °R.
Find the temperature, T_{2s}, at 100 psia for $s_2 = s_1$

$$T_{2s} = 180 + 20\frac{(1.4472-1.4439)}{(1.4592-1.4439)}$$

= 180+20(0.2157)=184.3F
h_{2s} = -614.16
+(0.2157)(-604.29+614.16)
= -612.03 BTU/lb
$-W_{IS} = h_{2s-h}$ = -612.03+652.68
= 40.65BTU/ibm
$-W_{act} = -W_{IS}/\eta_{IS}$
= 40.64/0.75
= 54.20

Find the actual outlet temperature from the outlet enthalpy by interpolation.

$h_2 = h_1 - W_{act} = -652.68 + 54.20$

$= -598.48$ Btu/lbm

$$T_2 = 200 + (20)\ \frac{(-598.48 + 604.29)}{(-594.23 + 604.29)}$$

$= 211.5$ °F

The power, pc, is now computed

pc = 50,000 x 54.2 / 2544 + 50 = 1115 hp «—

We can see from the similarity of the numbers in the computations that the enthalpy chart is based on Starling's tabulated data. The results are the same within the ability to read the graph.

Computer Simulation

Flow sheet simulator packages provide quick and simple means of solving the compressor design equations. The thermodynamics properties are used in exactly the same way as in the hand calculations above. The only difference is the source of the properties, which in the simulator is provided by ideal-gas properties (v, h, s) combined with equation-of-state-derived corrections for real-fluids properties.

Example 10-3. Solve the previous example using the OPSIM flow sheet simulator.

Solution: Shown below is a print out of the computer results.

Compressor Calculation:

Adiabatic Efficiency	= .7500
Mechanical Efficiency	= 1.0000
Delta P	= 80.0
Gas Horsepower	= 1078.2
Compression Ratio	= 5.000

Strm No	1	2
C3	1133.90	1133.90
MolHr	1133.90	1133.90
LbHr	50001.59	50001.59
T,Deg F	80.00	214.07
P,Psia	20.00	100.00
Mol Wt	44.10	44.10
Lb/CuFt	.1557	.6469
Btu/Hr	.71921E+07	.99362E+07
Btu/R-Hr	.66529E+05	.67569E+05
Frac. Vap	1.0000	1.0000

If 50 HP is added for bearing and seal losses, the horsepower is 1128, and the outlet temperature is 214.07 °F, which is excellent agreement with the thermodynamic tables.

Design Equations. The previously described isentropic and polytropic models generally provide reasonably good estimates.

Example 10-4. Use the Isentropic Model—Equation 10-14—to solve the previous Example 10-1.

Solution: The heat capacity of propane at an assumed average temperature of 150°F is:

Cp = 19.52 Btu/lbmole-°F (GPSA, 1987, p. 13-5)

Use the ideal-gas relation, $Cv = C_p - R$

Cv = 19.52 - 1.99

= 17.53 Btu/lbmole-F

k = Cp/Cv = 19.52 / 17.53 = 1.1135

Check the temperature assumption using Equation 10-18.

T_2 = 80 + (540)[(100/20)UI35/M135) - 1J/0.75

= 80 + 128 = 208 °F

T_{av} = (80 + 208)/2 + 144°F, which is close enough to the assumed value of 150.

But Zav = 0.96 (Starling's data) MW = 44.097 (GPSA, 1987, p. 23-2, Fig. 23-2).

$$-W_s = \frac{(.96)(1.986)(1.1135)(540)}{(44.097)(0.1135)} x[(100/20)^{.1135/1.1135} - 1]$$

= 40.82 Btu/lbm

$-W_{act}$= 40.82/0.75 = 54.4 Btu/lbm

This is close to the previous solutions, but not as accurate.

Example 10-5. Use the polytropic model to solve Example 10-1. Assume $\eta_p = 0.77$.

As in Example 10-3, estimate k = 1.1135 at assumed average temperature of 150°F.

$(n-l)/n = (k-l)/(k)(\eta_p)$

= (1.1135 - 1.0)/(1.1135) (.77) = 0.1324

$T_2 = T_1 (P_2/P_1)^{(n-1)} - 1$

= (80 + 460) $(100/20)^{0.1324}$

= 668°R = 208°F

$$-W_p = Z_{av} \frac{RNT_1}{MW(n-1)} [(p_2/p_1)^{(n-1)/n} - 1]$$

$$= (0.96) \frac{(1.986)}{(44.097)} \frac{1}{.1324} (540)[(100/20)^{.1324} - 1]$$

= 41.9 Btu/lbm

$-W_{act}$ = 41.9/0.77 = 54.4 Btu/lbm.

Reciprocating Compressor Cylinder Capacity

Obviously the piston cannot travel completely to the end of the cylinder at the finish of every discharge stroke. Some clearance volume, CV, is required and, therefore, some gas is left in the cylinder after the discharge stroke. As the piston travels away from the head end, this residual gas expands until the pressure is slightly less than that of the suction line. Then inlet gas can enter through the intake valves. The point is that the total piston displacement, PD (stroke length times cross-sectional area), is not available for throughput. Mathematically, the percent clearance, C (%) = 100 CV/PD. In turn volumetric efficiency, VE(%), is defined as the actual pumping capacity (at intake conditions) divided by the piston displacement. In theory, the volumetric efficiency can be calculated from the ideal cycle

$$VE = 100 - C (r1/k - 1)$$

To repeat, even in theory VE is less than 100% because the high pressure gas left in the cylinder at the end of the discharge stroke expands and so delays the opening of the suction valve. In practice, the VE is further reduced empirically to account for real gas behavior (Z 5^ 1), suction and discharge valve losses, etc.

$$VE = 96 - r - C [(Z_s / Z_d)(r^{1/k}) - 1]$$

Deduct an additional 5% for non-lubricated compressors and still another 4% for propane or other "heavy gases."

Example 10-6. A compressor cylinder has 100 cfm displacement (displacement volume times rpm) and a clearance of 10 percent. What is the volumetric efficiency for a compression ratio of 4.0 and k = 1.3? Assume ideal gas behavior.

Solution: $VE = 96 - 4 - 10[(4)^{1/1.3} -1]$

$= 96 - 4 - 10(1.905) = 73\ \%$ «-

The throughput is 73 cfm (approximately).

INTERNAL-COMBUSTION GAS ENGINES

Gas engines may be described and/or classified as follows

Combustion cycle:	two-stroke
	four-stroke
	Cylinder arrangement: vertical/upright
	Horizontal
	V-type
	opposed
Power impulse:	single acting
Speed:	low (below 700 rpm)
	medium (700-1500 rpm)
	high (above 1500 rpm)
Connection:	belt or separable
	direct drive
	integral

Gas engines are now discussed using the following topics: combustion cycles, engine description, and a brief comparison of two- and four-stroke engines.

Combustion Cycles

Figures 10-25 and 10-26 show schematic sketches of two-stroke and four-stroke cycle engines, respectively.

The conventional two-stroke engine is a compact unit consisting of gas engine and scavenging piston. Each cycle is one full revolution or two strokes. As shown in Figure 10-25, when the piston moves down (towards the crank end) in the power stroke it first uncovers (or opens) the exhaust port. This allows the exhaust gases to escape and lowers the pressure in the cylinder to atmospheric. When the intake port is opened (Fig. 10-26) air, previously compressed in the scavenging cylinder to about 5 psig, flows from the scavenging cylinder through the intake port into the cylinder. This air scavenges or expels the exhaust gases. When the piston moves upward (towards the head end), it closes the intake port first, then the exhaust port, and finally compresses the air. Then fuel gas is injected into the cylinder, and the compressed air/fuel-gas mixture is ignited near the end of the compression stroke. This combustion generates a high pressure (and temperature), thus forcing the piston down and completing the cycle.

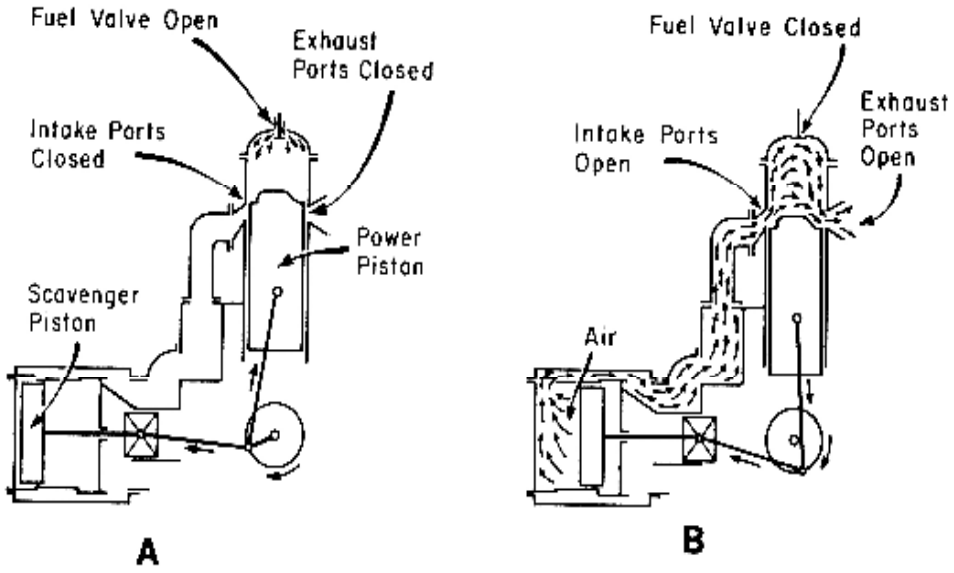


Figure10-25
Schematic sketch of a two-stroke cycle engine operation (courtesy PETEX, 1987.Copyright © 1987 by the University of Texas at Austin).

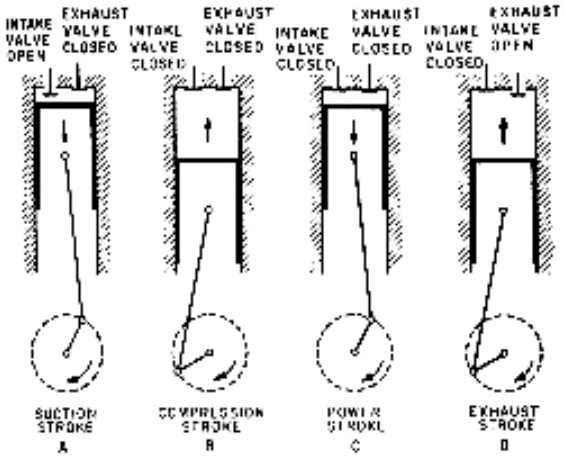


Figure 10-26
Schematic sketch of a four-stroke cycle engine operation (courtesy PETEX, 1987. Copyright © 1987 by the University of Texas at Austin).

In the four-stroke engine each cycle is four strokes or two full revolutions. As shown in Figure 10-26A, during suction stroke the piston moves downward, sucking air into the cylinder. As the piston starts to move upward (towards the head end) the intake valve closes and the air is compressed (Fig. 10-26B). During this compression-stroke the fuel gas is injected. Near the end of compression the fuel-gas/air mixture is ignited, and the resulting explosion forces the piston downward (toward the crank end), thus generating the power stroke (Fig. 10-26C). At the end of the power stroke,

the exhaust valve opens and the piston expels the exhaust gases (Fig. 10-26D) thus completing the cycle.
Figure 10-27 shows PV or indicator diagrams of the actual change in gas pressure and volume in the power-cylinder during the two- and four-stroke cycles.

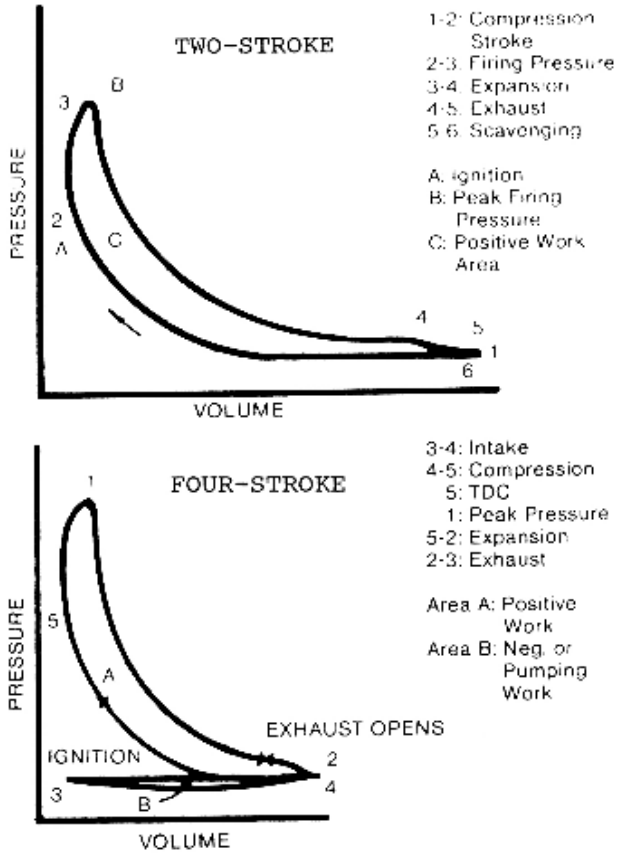


Figure 10-27
P-V diagrams (PMC/Beta Corp.).

Engine Description

Cylinder Arrangement. Power cylinders are usually arranged upright or vertically—in a V or opposed (Fig. 10-28).

Couplings. Internal-combustion gas engines are connected to the reciprocating compressors in one of three ways

1. Integral: A single crankshaft with multiple throws is used for both the power and compressor cylinders. Power cylinders are often vertical or V-type while the compressor cylinders are usually horizontal.
2. Direct Coupled: The driver and compressor crankshafts are coupled, therefore, speeds must be the same.

3. Belt Driven: Suitable for smaller units (less than 200 HP). Couplings also may be described as rigid or flexible.

Engine Starting. Standard L or V two-cycle engines are often started with compressed air at 250 psig by admitting air to the power cylinders and "rolling" the engine. Turbochargers are accelerated at start up by either starting air or hydro-mechanically.

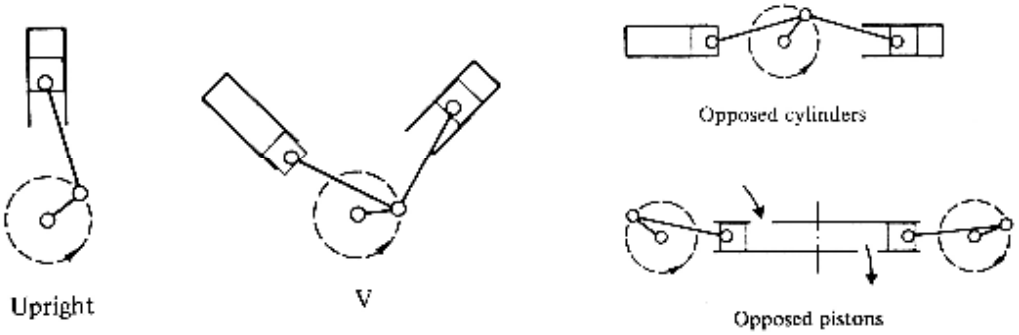


Figure 10-28
Common cylinder arrangements

Fuel-Gas System. Fuel gas should be fed through a pressure regulator into a header having at least 1 ft3 capacity per 100 Bhp of engine rating.

Timing Control. Automatic ignition timing control is standard, thus allowing the timing to be retarded during start up and gradually increased as the load is applied.

Air Filters. Clean air is essential for smooth operation; consult manufacturer for recommendations.

Ignition. Some of the more common ignition types are magneto-modified, dual-ignition, magneto-dual ignition, and pulse-generator dual ignition. In the pulse-generator type, low tension energy impulses are stepped up by a transformer mounted on the power cylinder head covers.

Lubricating Systems. Full-pressure lubrication is used. A crankshaft-driven, main oil, gear-type pump lubricates the engine and any mounted turbochargers (Fig. 10-29) while a mechanical lubricator supplies oil to the engine power cylinder walls, compressor cylinder walls, and compressor piston rod packing (Fig. 10-30). Each lubricating system can flood the running surfaces prior to start up.

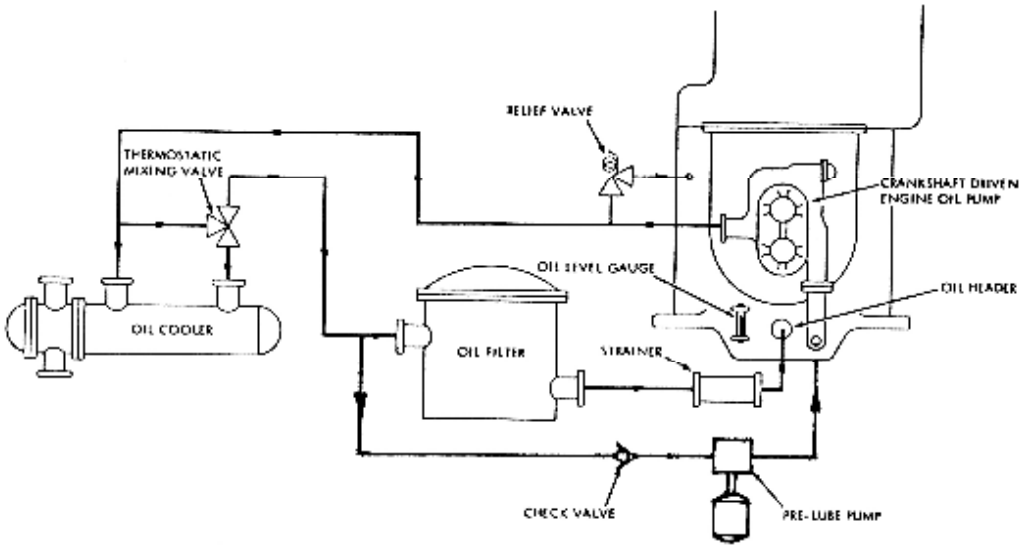


Figure 10-29
Typical engine lubricating system (Dresser-Clark; Berger and Anderson, 1980).

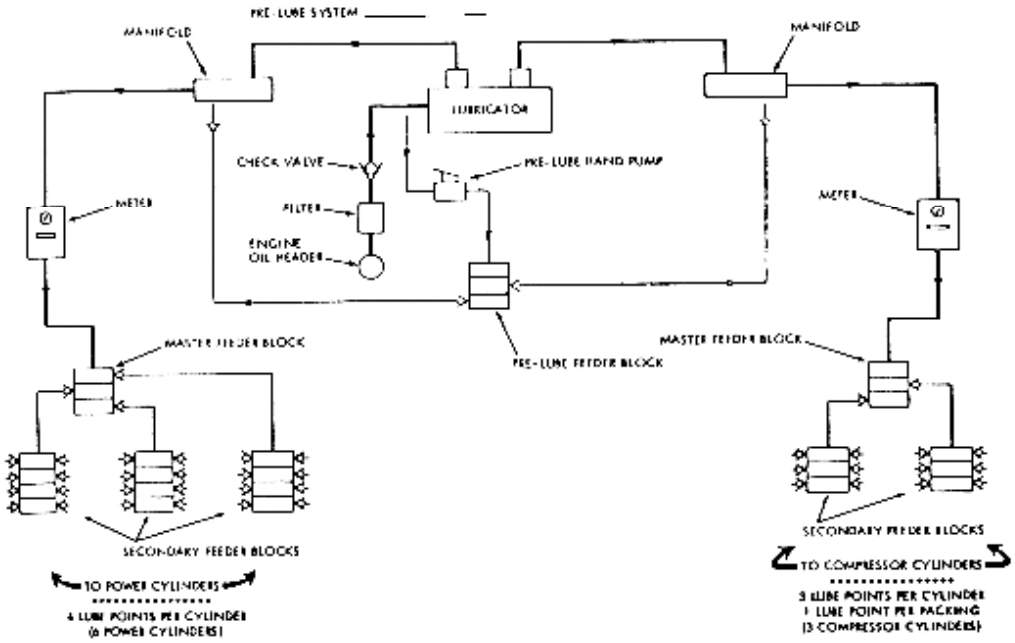


Figure 10-30
Typical cylinder and packing lubricating system (Dresser-Clark; Berger and Anderson, 1980).

Summary

Four-cycle gas engines are easier to start, are smoother running, consume less lubricating oil, and have higher thermal efficiencies than two-cycle engines. Two-cycle engines have no inlet valves, often no exhaust valves, require less displacement per Bhp; therefore, they are smaller, lighter, and cheaper. However, the differences are minor, especially in the turbocharged types. GPS A (1987, Section 15) provides further details.

GAS TURBINES

Gas turbines are self-contained power plants with many advantages over steam turbines (Molich, 1980):

1. No need for boilers, condensers, boiler-feed water, etc.
2. Substantially higher thermal efficiencies.
3. Can offer environmental benefits such as reduced blowdowns of cooling and boiler-feed waters.

General Description

Figure 10-31 shows the mechanical arrangement of a simple-cycle, single-shaft turbine. The centrifugal or axial compressor supplies air to the combuster where the fuel is burned at constant pressure. Then the resulting power gas is expanded through the turbine. About two-thirds of the turbine output is used to drive the inlet air compressor, and the remaining one-third is converted to useful output shaft power.

Only about 20% of the injected and compressed air is used for combustion. The remaining 80% cools the combustion chamber to 1400°F for heavy-duty types and to 2200°F for some high-performance aircraft derivatives.

Figures 10-31b,c,d illustrate how the gas engine configuration can be modified to improve efficiency. Two-or three-shaft arrangements allow each shaft to rotate at the correct speed for maximum efficiency.

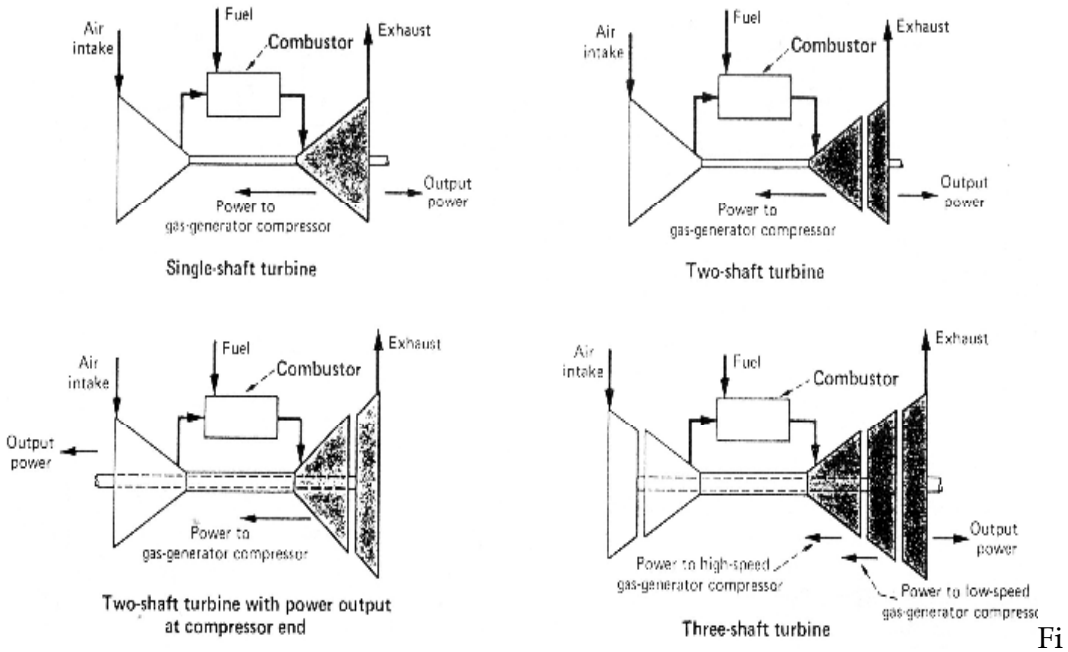


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Figure 10-31
Gas turbine shaft configurations (Molich, 1980, Vol. 95, No. 10, 22-26, 1988).

The efficiency of the simple cycle (Fig. 10-31) can be improved by three modifications. First is regenerative heating which uses the hot turbine exhaust gases to preheat the compressor discharge air before it enters the combustor. Second, the compressor work can be reduced by intercooling. Third, the turbine power output can be increased by interstage reheating. While Figure 10-32 shows all three modifications, regenerative heating is often used alone. The closed-cycle gas turbine is finding increased used—see Molich (1980) for a good description. Figure 10-33 depicts a Mars turbine engine.

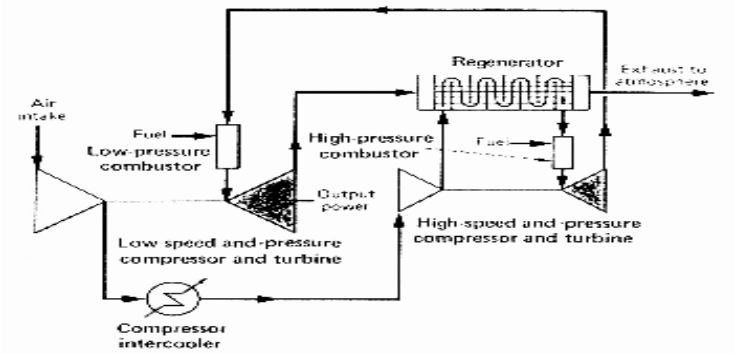


Figure 10-32
Modifications to improve cycle efficiency (Molich, 1980, Vol. 95, No. 10, 22-26, 1988).

Gas Turbine Cycles

Figure 10-34 shows the basic gas turbine cycle or Brayton cycle. The ideal Brayt on cycle assumes:

1. Air is compressed isentropically from 1 to 2.
2. Combustion occurs at constant pressure: 2 to 3.
3. Isentropic expansion occurs in turbine: 3 to 4.

Line 4to1does not exist in practice as the exhaust gases leave the engine. Nevertheless the thermal efficiency, $\eta_{thermal}$ of this cycle is (Anson, p. 3-6)

$$\eta_{thermal} = \frac{useful work}{heat supplied} = 1-(1/r)^{(k-1)/k}$$

In practice the efficiencies of the compressor, r\c, and turbine, TIT. must be considered. Now (Anson, p. 3-7)

$$\eta_{thermal} = (\eta_c \eta_T K-1)/\eta_C (K-1) a$$

Where K = ratio of turbine to compressor ideal works.
Sorensen (1983) calculated how the thermal efficiency of gas turbine varies with the pressure ratio, r, and the maximum temperature, T3. Figure 10-35 shows the results for representative values of inlet air temperature, T,, and compressor and turbine efficiencies. Combustion, heat, pressure, mechanical, and turbine losses are also included (Sorensen, 1983).

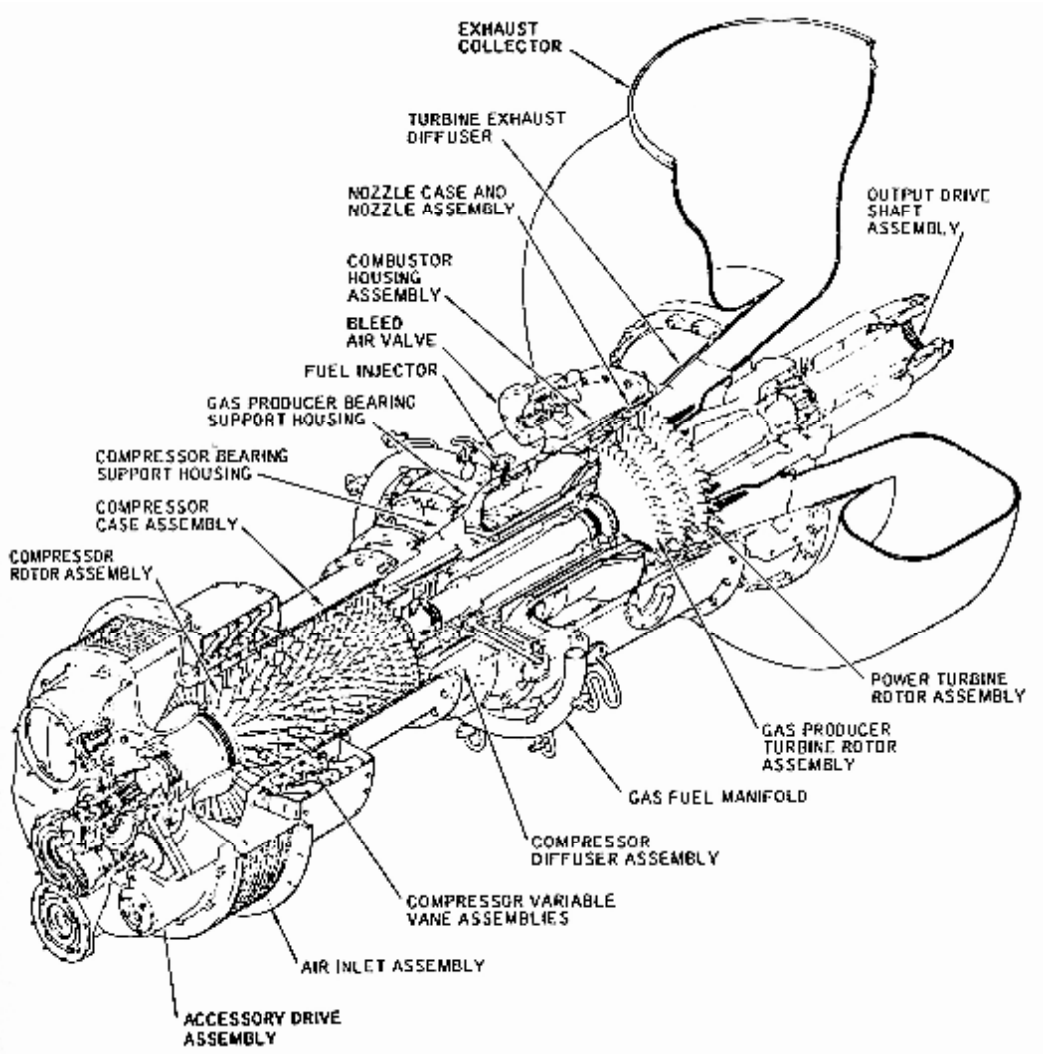


Figure 10-33
Mars turbine engine (Solar Turbines, Inc., 1989).

Materials of Construction

Compressor and turbine efficiencies have reached about 90%, and major efficiency improvements are unlikely.

However, maximum safe firing temperature for turbines are increasing continuously. There are three approaches (Molich, 1980):

1. Use materials with high creep-rupture strength for highly stressed rotating components such as first-stage turbine buckets and disks, first-stage turbine nozzles, combustor liners, and transition pieces. Nickel and cobalt based alloys are used.

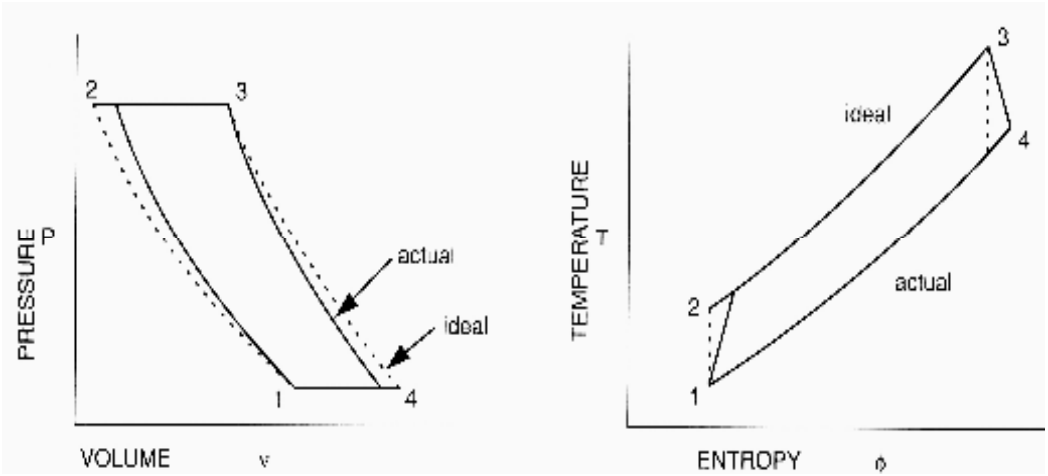


Figure 10-34
Brayton Cycle (Anson).

2. Protective thermal-barrier coatings for high-temperature service; for example, aluminum oxide, tungsten and chromium carbides, alloyed with Ni, Cr, Pt.
3. Air cooling for turbine nozzles, buckets and disks can keep components 250-500°F below hot-gas temperature.

Gas Turbine Types

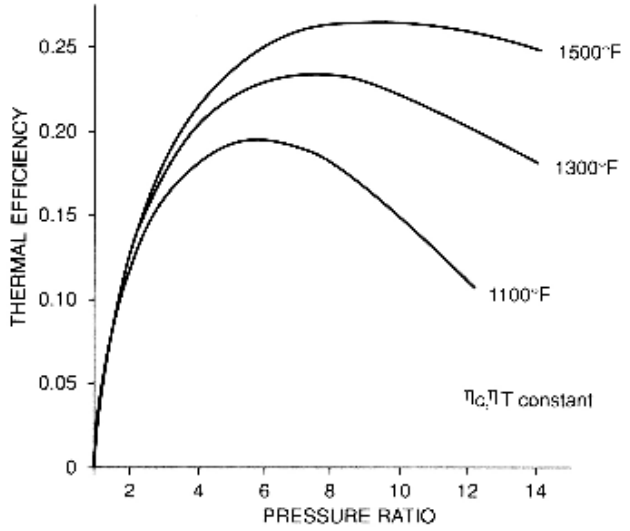


Figure 10-35
Thermal efficiency of gas turbine (Sorensen, 1983).

The two types of gas turbines are the heavy duty and the aircraft derivative (Molich, 1980). The heavy duty gas turbine is suitable for normal industrial uses without space or weight limitations. The turbine is either single- or two-Shaft type and the compressor, turbine blades and buckets, and turbine nozzles are of heavy design. Moderate compression ratios (6-9) and power gas temperatures (1700-1900°F) are selected for long-time (100000 hour) continuous duty between inspection and maintenance. Shaft bearings are of conventional design—sleeve or tilting pad for radial bearings and tapered land or multiple segment for thrust bearings—arranged for operation with a pressure lubrication system common to the gas turbine and driven equipment. The turbine, lubrication system, auxiliary systems, and instrumentation are stipulated by API Standards 614 and 616.

The aircraft-derivative gas turbine is essentially an aircraft jet engine with the jet nozzle replaced by a power turbine. Four advantages are:

1. Sophisticated aircraft technology and associated research and laboratory facilities.
2. Mass-production techniques and quality-control methods used by the aircraft industry.
3. The service centers for aircraft jets, with their rigorous certification requirements, spare-parts inventories, and test facilities are available.
4. Fast changeout of the entire gas generator element, typically 12 hours. Overhaul need not be done "in situ."

The aircraft-derivative gas turbine is inherently of either the two- or three-shaft configuration. The regenerative cycle cannot be applied for this turbine type. The power turbine and the gas generator (the jet engine) are separate entities, with no mechanical linkage. This separation also applies to the auxiliary systems. The power turbine is of heavy design and shares its conventional accessories, instrumentation, and lubrication system with the driven equipment. Gas generator exhibits lighter weight and compactness imposed by aircraft design, extensive use of antifriction bearings; special lubrication systems using synthetic, nonflammable lube oil; and hydraulic-type accessory and electronic-hydraulic instrumentation. When aircraft units are used in industry, the firing temperature and power output are reduced considerably to increase life span and time between overhauls.

Environmental Impact

Two distinct problems are noise and stack emissions. To meet the normal noise criterion of 85 to 90 dBa at a 3-foot distance, inlet and exhaust silencers together with acoustic enclosures are required. Noise-abatement components are now of heavy structural design and often of the free-standing type. Outer walls may be fabricated of 3/i6- to !/4-in.-thick steel plates and the enclosures sound insulated. High

temperature corrosion-resistant materials are used for the exhaust stack. The sound-absorbing materials may be mineral or glass fiber.

Normal stack pollutants, such as sulfur dioxide, carbon monoxide, and particulates are controlled by fuel specifications and turbine combustor design. Nitrogen oxides, NOx, are much more troublesome; however, NOx emission can be controlled by injecting water of boiler-feed quality into the combustors.
In the U.S., EPA permits may be required before use of new fuel-burning engines. Obtaining these permits can take months.

Fuels

Gas turbines can use many fuels: liquid fuels include light distillates, diesel oil, gas oil, bunker C, and crude oil. Gaseous fuels include natural gas, propane, butane, refinery gas, hydrogen, carbon monoxide, and coke-oven gas.
Fuel contaminants such as Na, K, V, or S can and do cause problems. Any sulfur in the fuel forms SO2 or SO3 when burned; while metals such as Na, K, and V, can form low melting eutectics with some brick refractories. These problems are discussed in Volume 2 in the chapters on fired heaters and energy conservation.

Operation and Controls

The standard control system automatically starts, runs, monitors, and stops the unit. In addition, the controller monitors the unit and protects it from abnormal operation. The automatic start-up sequence occurs in one minute. First the starter motor—an electric, hydraulic, pneumatic, or gas expansion motor—spins the machine for about half

A minute. This purges the turbine and exhaust systems. Then fuel is introduced and fired by the ignitor. Finally the machine is loaded at unit speed.

Gas turbine operation requires careful and extensive monitoring as illustrated in Table 10-1.
s
Turbine performance deteriorates as the turbine and compressor blades become fouled. Clean fuel will reduce turbine fouling; however, regardless of the level of air filtration, the air compressor blades will foul. Abrasive cleaning by adding nut shells or rice to the air intake is often performed regularly—maybe once a week. Special nozzles are required for abrasive cleaning of the turbine (Molich, 1980).

Washing with hot water, detergent, or solvent is required if the compressor has digested oil or if ash is deposited on the hot-gas parts (fuel nozzles, combustors, transit pieces, turbine nozzles, and buckets) (Molich, 1980).

Table 10-1 also summarizes a typical alarm and shutdown system and provides some insight into the sophistication of the control system.

PREVENTIVE MAINTENANCE

Compressors are generally the most expensive gas-processing equipment to buy and operate. Good maintenance is essential. The objectives of any maintenance program are to:

1. Ensure absolute safety for all personnel.
2. Keep equipment running reliably.
3. Minimize costs.

Meeting the above goals is most challenging. Cost control in particular is getting harder every year. Accordingly, alternate maintenance philosophies are discussed first and then the essential ingredients for a workable preventive maintenance program are reviewed.

Maintenance Philosophies

A maintenance program can follow three philosophies:

1. The "poor boy" marginal approach that believes fervently in "run it 'til it quits" or "if it ain't broke, don't fix it."
2. The fancy "whole-life insurance policy," i.e., regular complete overhauls on a previously defined schedule.
3. Planned preventive maintenance designed to provide adequate repair and monitoring services at minimum cost.

The "poor boy" marginal approach is very costly in that, without adequate inspections, breakdowns occur

Table 10-1 Alarm and Shutdown Indication (Solar Turbines Inc., 1989)

Indication	Alarm (Indication Only)	Shutdown* (Indication and Shutdown)	Supplied With
Engine System		L	Basic set
High Engine Temperature (T_7)		NL	Basic set
Fail to Crank		NL	Basic set
Fail to Start		NL	Basic set
Ignition Failure		NL	Basic set
Overspeed Power Turbine		NL	Basic set
Lube System			
High Oil Temperature		NL	Basic set
Low Oil Pressure		L	Basic set
Low Prelube Oil Pressure		L	Basic set
Low Oil Level		L	Basic set
Fuel System			
Low Gas Fuel Pressure		NL	Basic set
Compressor Systems			
Low Seal Oil ΔP		L	Basic set
Fail to Load		NL	Basic set
High Discharge Gas Pressure		NL	Optional
High Discharge Gas Temperature		NL	Basic set
Compressor Surge		NL	Optional
Yard Valve Sequence Failure		L	Basic set
Low Suction Gas Pressure		NL	Optional
Ancillary Systems			
Low Battery Voltage		NL	Basic set
Inlet Air Filter—High AP	X	NL	Inlet air filter
Inlet Air Filter—Blow-In Door Open	X	NL	Inlet air filter (with blow-in door)
Inlet Air Filter—Blower Motor Failure	X		Inlet air filters that include a scavenge fan
High Enclosure Temperature	X		Enclosure (optional)
Low Enclosure Pressure	X		Pressurized enclosures (desert application)
High Gas Level (enclosure)		L	Gas detection system (optional)
Gas Monitor Failure (enclosure)	X		Gas detection system (optional)
Fire System Locked out	X		Fire protection system (optional)
Fire System Discharged	X		Fire protection system (optional)
Instrument Air System Failure	X		Surge control system, etc.
High Fuel Gas Scrubber Level	X		Requires input signal from external equipment

High Fuel Gas Filter – AP	X		Requires input signal from external equipment
High Gas Cooler Discharge	X		Requires input signal from external equipment
Temperature			
High Scrubber (1) Level		NL	Requires input signal from external equipment
High Scrubber (2) Level		NL	Requires input signal from external equipment
Low Scrubber (1) Level		NL	Requires input signal from external equipment
Low Scrubber (2) Level		NL	Requires input signal from external equipment
Gas Cooler (1) Vibration		NL	Requires input signal from external equipment
Gas cooler (2) Vibration		NL	Requires input signal from external equipment

* L = Lockout, NL = Nonlockout

Unexpectedly. This "reaction maintenance" results in unscheduled repairs, usually performed under adverse conditions. As predicted by Murphy's law, breakdowns always seem to occur at particularly inconvenient times that result in large losses of production. And catastrophic failures of compressors often cost millions of dollars to repair (Dodd, 1984).

The "whole-life" method is practiced in regulated industries, e.g., airline companies. In maximizing passenger

Safety, engines are overhauled and parts replaced whether needed or not. This overhauling at specified intervals regardless of condition not only wastes time by disassembling good machinery but reassembly often generates unnecessary start-up programs and can increase wear.

Planned preventive maintenance is the sensible approach. The goals are to monitor equipment so that repairs are performed only when needed. This "condition maintenance
Only" plans to run the machine its last mile without wrecking it (Dodd, 1984). James (1982) defines good maintenance by two rules:

1. If a machine is running well by analytical check, leave it alone.
2. If a machine is not running well, fix only what needs to be fixed.

Any satisfactory preventive maintenance program must involve the following:
1. All applicable governmental regulations must be followed, and industry codes

(e.g., API's RPs) should be considered carefully.
2. A good lubrication system.
3. Reliable monitoring and shutdown systems.
4. Thorough on-line inspections.
5. Careful downtime inspections and servicing.
6. Adequate diagnostic testing.
7. Adequate record keeping.

These seven items are now discussed. Lane et al., (1988) describe in detail the successful implementation of such a preventive maintenance program in a gas plant.

Regulations and Codes

Government regulations and industry codes are designed to protect employees. Ignoring governmental regulations is not only illegal but can result in very expensive litigation and employee compensation. On the other hand API Recommended Practices are just that—recommendations— and should aid, and not control engineering judgment.

Lubrication System

A sound lubrication program is absolutely essential for a successful preventive maintenance program. Manufacturers' recommendations and/or company specifications should be followed closely. High-performance specialty lubricants may not be suitable for other applications and, even if suitable, might not be worth the cost.

When monitoring lubrication remember that too much oil is almost as bad as too little. Excess oil can cause hydroplaning, damage cylinder walls and pistons, and form excessive deposits. Dirt, abrasives, water, and diluents must be kept out of the oil. Filters should be checked regularly and changed when required. Remember that dirty gas is a frequent source of oil contamination. Lube oil must be tested regularly and replaced promptly when needed.

Monitoring and Shutdown Systems

To be satisfactory, a monitoring system must give adequate advance warning of impending trouble and shutdown the Machine thereby avoiding (or at least minimizing) damages from catastrophic failures.

As a minimum the following should be monitored: engine, compressor and fan vibration, cooling water temperature and pressure, high and low gas pressures, scrubber high level, high gas discharge temperature, high engine exhaust temperature, engine overspeed, turbocharger low discharge air pressure, and lubricating oil pressure and flow. On the other side of the spectrum is the API 670 machinery protection system that continuously monitors vibration, axial position,

and thrust bearing temperatures. Dodd (1984) outlines the success of API 670 in avoiding catastrophic failure of major turbomachines.
Safety shutdowns should protect against random failure and/or operator error. To be successful a safety shutdown system needs:

1. Good design and reliable components
2. Regular component testing program (follow codes such as OCS, MMS, and API RPs)
3. On-line testing capability
4. Strict policy against bypassing safeguards.

Daily On-Line Inspections

On-line inspection programs need: reliable instrumentation; comprehensive check list; good, concise operating procedures; thorough follow-up on suspected problems; and thorough operator training. Pressure and temperature gauges should be checked daily and replaced immediately when defective. Daily inspections should include: knocks and unusual noises; excessive vibration; compressor suction and discharge pressures and temperatures for all stages; engine speed; lubrication system pressures, flows, levels, temperatures, and filter pressure drop; crankcase oil appearance; compressor rod packing vent for excessive gas leakage; compressor rod packing oil drain for discolored oil; engine manifold pressure; turbocharger speed and vibration; jacket water temperatures, flows, pressures, and levels; oil and water leaks; engine exhaust temperatures; governor oil level; scavenging air pressure; fan belts; ignition wiring; fuel gas pressure; fuel gas scrubber liquids; suction and discharge scrubber liquids; starting air receiver water level; loose anchor bolts, crosshead bolts, head bolts, and flange bolts; housekeeping; and others as recommended by manufacturers or learned from practical experience.

Downtime Inspections and Servicing

Early diagnosis of problems is the key when an engine is shut down for routine servicing such as oil change, oil filter change, spark plug or magneto change, etc., this is a "free" opportunity to check for potential problems. Some of the following checks should be made: compressor rods for signs of excessive wear; compressor cylinder walls for unusual wear by pulling top valve from outboard end of compressor cylinder and spotting the piston at crank end (a light and inspection mirror will help and all safety precautions should be taken); and fan belts for wear. If oil is being changed, remove cover plate from engine crankcase and check the internals—look for metal cuttings and sludge deposits in the crankcase, and look at all bearing caps for signs of overheating, loose bolts, and other evidence of possible trouble. If filters are being changed, check old filter elements for metal cuttings. Check combustion and scavenging air filters and clean when needed. With crankcase open, check power cylinder liners for scuffing and wear by spotting each piston at

TDC (take all necessary safety precautions). If spark plugs are being changed, run a compression check on each power cylinder. Check valve tappet clearance; cylinder jackets or engine and compressor for scaling, corrosion, etc.; oil and jacket water coolers for scale, corrosion, leaks, etc.; and any other internal inspection points that may be accessible without excessive dismantling.

Diagnostic Testing

This field has been revolutionized in the last 10 years. Some of the instruments in current use include the following items.

Portable Computer with CRT (cathode ray tube with video display screen) and keyboard. This unit can be "plugged in" and will measure and report engine pressures and temperatures, movement of pistons, and time of revolutions of crankshaft. The goal is improving and maintaining fuel efficiency.

Engine/Compressor Analyzers including an oscilloscope, for reciprocating engines and four channels, inspect (PMC Beta) ignition voltage; vibrations; firing pressures; and timing. Obtainable information includes: power cylinder horsepower readout; compressor cylinder horsepower readout; engine speed readout; compressor volumetric efficiency; compressor cylinder pressure-volume oscilloscope display; power cylinder pressure-time oscilloscope display; vibration-time oscilloscope display; and ignition-time oscilloscope display. With this information, the following analyses of the engine/ compressor performance can be made: power losses, bad engine and compressor valves, defective rings and bearings, scored parts, carbon deposits in parts, scored cylinder walls, piston slap, and faulty ignition components.
A Fiber Optic f robe—flexible, periscope-type instrument—actually bends light to see around corners and over obstacles deep within the machines. It can inspect turbine blades and buckets as clearly as if the engine were disassembled and, it can record views by color photography and/or television.

An Ultrasonic Detector complete with display screen can detect cracked bolts by "shooting" an ultrasonic wave down bolt as the engine runs. Another type of ultrasonic detector—a cube-shaped listening device or stethescope— checks for valve leakage when the engine is running.

Spectroscopic Analysis of Lubricating Oils is a quick, simple, and low-cost way of checking engine wear. Wear occurs as ultra-fine metal particles which are too small to be filtered out of lube oil. For example, Pb, Cu, Sn, Al, Ag, indicate bearing wear; Fe, Cr, Ni indicate liner or piston ring wear; Si is a contaminant from the air intake system; B, Na come from the cooling water system; P, Zn, Ca, Ba are related to lube oil additives. Oil analysis complements electronic detectors, requires no special tools, and can be subcontracted to reliable laboratories. Monthly sampling is recommended.

Ferrography—A direct reading ferrograph (Fig. 10-36a) measures the relative number of small (1-2 (μm) and large (> 5 (μm) metal particles in the lube oil. Figures 10-36b and 10-36c show pictures of wear particles while Figure 10-36d shows how the size and number of metallic wear particles vary with time. Laskowski (1984) documents the benefits of using ferrography.

Optical Alignment Devices—Instrument looks like a telescope, is positioned on a tripod, and is an ultra-accurate engineering level that "shoots" levels when the engine is reassembled after an overhaul. Use with an optical micrometer. Can set bearings within 0.001 in. of desired elevation; this prevents wear, especially crankshaft deflection.

Troubleshooting

The GPS A (1987) Engineering Data Book (Section 13) provides excellent one-page troubleshooting charts for reciprocating (p. 13-21) and centrifugal (p. 13-39) compressors, respectively.

Record Keeping

• Without adequate record keeping it is impossible to assess performance properly or to establish the benefits of preventive maintenance. However, excessive record keeping wastes money and hurts employee morale. Broyles (1978) discusses this dilemma and recommends the following parameters for monitoring monthly compression performance:

1. Percent of available HP that is used.
2. Specific fuel consumption
3. Downtime
4. Volume of gas flared

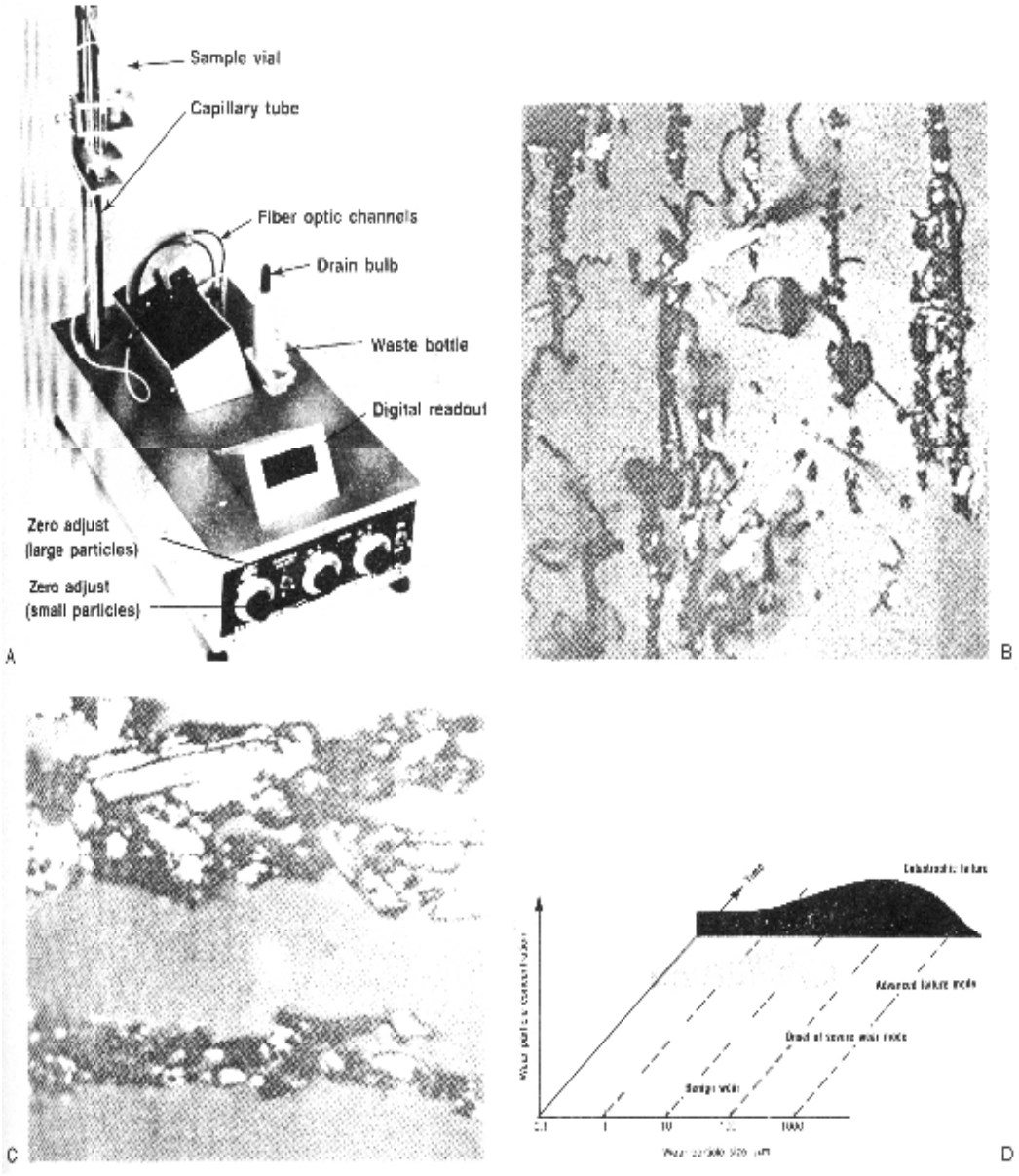


Figure 10-36

DR Ferrography (Lakowski, 1984).

SELECTION OF COMPRESSORS AND DRIVERS

Reciprocating compressors operate at relatively low speeds, say 200-800 rpm. The most common driver for small reciprocating compressors is the electrical motor. The speed of the motor is reduced by means of pulley and belt drive. The larger pulley, or sheave, is mounted on the compressor shaft and is driven by the smaller pulley on the motor shaft. In this arrangement there is no speed control possible, except perhaps by a factor of two, since AC induction motors are inherently constant in speed. Direct coupling of the motor to the compressor or connection through a gear train are less-used alternatives.

Medium- to large-sized reciprocating compressors are driven by gas engine, steam turbine, or electric motor depending on convenience and economics. In the field Compression of natural gas the gas engine is almost always used. In a gas plant, steam may be used due to its availability. Connection of driver to compressor may be by coupling or gear train. In large units, the driver and compressor may be integral (both operating off the same crankshaft).

The high speed of centrifugal compressors limits choice of drivers somewhat. In field applications, the most common driver is the aircraft-derivative gas turbine. Gas turbines are either directly-connected or geared to the compressor. In the plant, steam turbines and electrical motors are common. Power recovery is sometimes obtained by using expansion turbines (expanders) with high-pressure process gas streams. Turbine drives have the advantage of easy speed control. Electrical motors are generally constant-speed devices. Variable-speed, variable-frequency AC motors can be used, but they are relatively expensive.

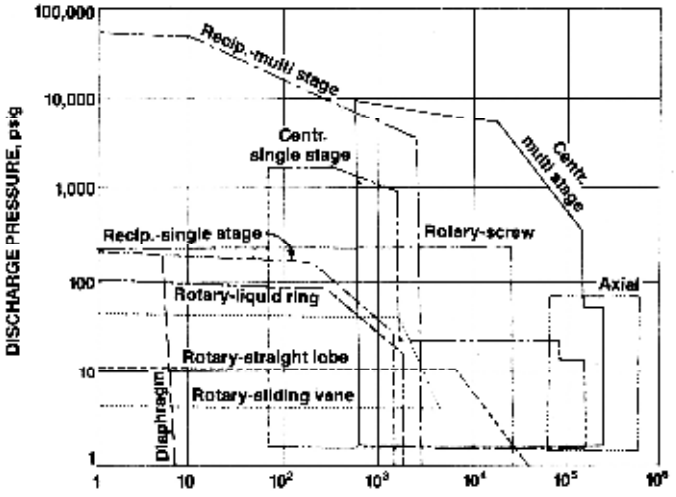


Figure 10-37
Compressor selection guidelines (GPSA, 1987, p.13.3)

Figure 10-37 (GPSA, 1987, p. 13-3) depicts the usual choice of compressor type as a function of discharge pressure and inlet flow (ACFM at inlet flowing pressure and temperature). Rotary compressors are generally used in special-purpose applications, such as wet or corrosive gases, vacuum applications, and the like. Jet ejectors are used almost exclusively in vacuum applications. The main choice lies between recips and centrifugals.

Reciprocating compressors can handle much lower flow rates per frame than centrifugal compressors—two orders of magnitude lower. Both types are competitive to 10000 psig discharge pressure, which covers the vast majority of applications. Note there is a definite lower limit to the flow rate of centrifugals. Axial compressors are seen to be confined to very high-flow, low-discharge-pressure applications.

Since reciprocating compressors are high-maintenance devices, it is common practice to specify two or more in parallel for a given service. Centrifugal compressors, on the other hand, are generally sized for the maximum expected load and a single machine is specified. Given these facts, it is clear that the overlap in service is broader than Figure 10-37 might imply.

Matching the driver to the compressor is, obviously, most important. Gas engines and reciprocating compressors are readily compatible because of their mutual reciprocating action and relatively slow (1000 rpm or less) speed. Offshore, high-speed gas turbines with double reduction gears are used to drive recip compressors. Gas turbines and axial compressors also pair due to their mutual high-speed (greater than 5000 rpm) rotary motion.

Centrifugal compressors operate best with high throughputs and relatively stable volume flows, suction, and discharge pressures. Multistage centrifugal compressors can generally achieve about the same compression ratios as single-stage recips. They have lower fuel efficiencies than recips. They also have fe\vex_jnoving parts and, therefore, lower maintenance and lubrication costs. Centrifugal compressors have balanced, steady forces; low initial installation costs; and simple auxiliary equipment.

Reciprocating compressors are suitable for low or high compression ratios at varying rates under varying suction and discharge pressures with a wide range of through put volumes. Recips have higher overall thermal efficiencies, many moving parts, and higher maintenance.

Economic Comparison

Shaw (1980) claimed that both reciprocating and centrifugal units are capable of meeting virtually any application. Shaw further argued that the selection choice should be made on economics, i.e., total cost of sejryjge. Accordingly, Shaw estimated capital costs, annual operation, maintenance costs, and fuel efficiencies and projected cost escalations. He computed the total cost (present worth) for 20 years of gas-transmission service. This analysis involved the following assumptions:

1. 95% load and 95% use factors.
2. Working capital and land will amount to 37.5% first-year maintenance and operation costs.
3. Investment financed 75% by bonds (12% interest) and 25% equity.
4. 20-year life and 15% DCF rate of return.

Shaw's computations show that the recips' higher first-year cost and higher operation and maintenance are very nearly balanced by {heir superior, fuel efficiency. This apparent economic equality probably explains why so often the choice is made primarily on prior experience and/or personnel preference.

Conclusions

The optimum choice requires careful and complete study of all relevant plant, site, project, process, and machinery parameters. Nevertheless the following recommendations have much merit.

1. Low-speed integral reciprocating compressors are best when:
    a. High fuel cost
    b. No waste heat required
    c. Large horsepower increments are required
    d. Maximum operating flexibility is needed
    e. Block mounted installation is OK
    f. Long delivery/construction schedule OK
    g. Long project life
    h. Very high pressures involved
    i. Heavy molecular weight gases are involved.

2. High-speed separable recips are best when:
    a. Medium fuel gas cost
    b. Little or no waste heat is required
    c. Minimum initial cost is required
    d. 4000 BHP or less horsepower increments required
    e. Short project life is involved

f. Relocation and/or conversion may be required
g. Minimum shipping/construction schedules.

3. Gas turbine/centrifugal compressors are best when:
   a. Good process fit
   b. Waste heat is required
   c. Minimal foundation requirements are necessary
   d. Light weight desirable
   e. Maximum reliability required
   f. Short—medium delivery/construction schedule required
   g. Minimized fieldwork is required
   h. Low fuel gas cost.

Review Questions

1. Compare reciprocating, screw, and centrifugal compressors. Include the following considerations:
   - equipment size, weight, and cost
   - maintenance
   - fuel efficiency
   - choice of driver
   - Overall economics.

2. List five methods for sizing a compressor (see Examples 10-1 through 10-5). Compare these procedures. Include:
   - ease of calculation
   - data required
   - accuracy.

3. List and compare three maintenance philosophies.

4. What are the key factors for any good maintenance program?

Problems

1. Estimate the horsepower required to compress 2 MMscfd of gas measured at 14.65 psia and 60°F.
   Intake conditions are 36 psig and 100°F.
   Discharge pressure is 436 psig.
   Gas gravity is 0.55.
   Atmospheric pressure is 14.2 psia at 1000 ft elevation.
   Assume isentropic efficiency = 85%.

2. A three-stage reciprocating compressor compresses 4.88 MMscfd of gas from 135 psia and 110°F to 1150 psia.
   a. If the gas specific gravity is 0.863, estimate the horsepower required.
   b. Estimate how the compression horsepower varies if: The compressor suction pressure varies by 15%. The suction temperature varies by 30°F.

   The required discharge pressure decreases by 40%.
   The required discharge pressure increases by 20%.
   The pressure drops across the two intercoolers
   Increase from 5 psi to 20 psi.
   The gas specific gravity increases to 1.
   The k value increases from 1.2 to 1.3.
   The compressor speed increases 11% to 1000 rpm. Mak (1987) discusses the effects of these changes in detail.

Nomenclature

Bhp = brake horsepower
C = clearance, %
C_p = specific heat at constant pressure, Btu/lb °F
C_v = specific heat at constant volume, Btu/lb °F
CG = correction factor for gas gravity, Equation10-30
CHS = correction factor for high speed, Equation10-30
CLIP = correction factor for low intake P, Equation10-30
CV = clearance volume
F = stage factor, Equation 10-27
Ghp = gas horsepower
H = head, ft Ibf/ Ibm or "ft"
h = enthalpy, Btu/lb, kJ/kg
hp = horsepower
k = ratio of Cp /Cv
m = mass flowrate, Ib/hr, kg/s
MF = multiplication factor, Equation 10-27
MMcfd = million cubic feet at 14.4 psia and T_s
MW = molecular weight, Ib/lb mol
N = number of compression stages
n = poly tropic coefficient, Equation 10-20
P = absolute pressure, psia, kPa
p = power, hp, ft Ibf/sec, kW R = ideal gas constant
r = compression ratio, Pd /Ps (absolute pressures)
T = absolute temperature
VE = volumetric efficiency, %
W = compression work, Btu/lb, kJ/kg
Z = compressibility factor

Greek

η = efficiency (theoretical/actual)

Subscripts

Act = refers to actual work
av = refers to average property
c = refers to compression
d = discharge condition
g = refers to gas
is = denotes isentropic change
L = denotes standard sales conditions
p = denotes a polytropic change
s = suction condition
T = refers to turbine

Thermal = refers to thermal efficiency
1 = suction or inlet condition
2 = discharge or exit condition
2s = discharge condition, isentropic path

MODULE 3:

Gas Dehydration

GAS DEHYDRATION

INTRODUCTION

Natural gas is usually in contact with water in the producing reservoir, and comes to the wellhead saturated with all the water vapour it can hold. It may either be dehydrated at the wellhead or brought into the plant wet (using some means to prevent hydrate problems. If an amine or other aqueous sweetening process is used in the plant, the sweetened gas will again be saturated with water.

Removing most of the water vapor from the gas is required by most gas sales contracts. because it prevents hydrates from forming when the- gas is cooled in the transmission and distribution systems and prevents water vapor from condensing and creating a corrosion problem. Dehydration also increases line capacity marginally.

Most sales contracts in the southern United States call for reducing the water content in the gas to less than 7 lb/MMscf. In colder climates, sales requirements of 3 to 5 lb/MMscf are common,

GAS HYDRATES AND HYDRATE CONTROL

Gas hydrates are rather complicated compounds of hydrocarbons, water and other substances such as hydrogen sulphide or carbon dioxide which form granular solids, very similar to ice, under certain conditions of temperature and pressure. There is some doubt whether hydrates can form if the water is mixed with liquid hydrocarbons, however, it should be realized that if water is present in the liquids, it very probably will also be present in the gaseous phase through the equilibrium conditions which will be reached in any pipeline system.

Individual components alone, or mixtures of the lighter components of natural gas, will form hydrates over a wide range of temperatures, depending upon the system pressure in the case of individual components, and composition also in the case of mixtures. These temperatures are considerably above the normal freezing point of water and may go higher for a sweet gas. The effect of heavier components is to increase the temperature at which hydrates will form for any given pressure. Normally, hydrates of components heavier than butane are not considered to occur. In all cases, free water in the system is required to form hydrates.

The presence of hydrogen sulphide greatly increases the temperature at which hydrates will form. Sour gases may have hydrate temperatures into the 90°F temperature range under high pressures.

WATER DEW POINT

When a gas has absorbed the volume of its water holding capacity at specific pressure and temperature , it is said to be saturated up to that point .
Any additional water added at the saturation point will not vaporise , but will fall out as free liquid.

Also at this point , if the pressure is increased and the temperature decreased the capacity of the gas to hold water will decrease and some water vapor will condense and drop out.

This point is known as the water dew point which may be defined as the temperature at which the natural gas is saturated with water vapor at a given pressure.

At the dew point, natural gas is in equilibrium with liquid water.

DEW POINT DEPRESSION

The difference between the dew point temperature of water in saturated gas stream and the stream after it has been dehydrated is known as the dew point depression .
To illustrate the concept of dew point depression, suppose that the natural gas at 500 psia, and 60° F , at the saturation point contains 30 Ibm of water per million cubic feet.

The dew point of this gas is .60°F. Suppose this natural gas is going to be transported in pipeline at 20° F. The saturation point will then be 7 Ibm of water /MMCF. The original 30 Ibm of water, if left in the gas, will exist in the from of 7 Ibm of water vapor and 23 Ibm of free water per million cf, if the pressure remains the same.

This free water is a potential source of hydrates to form and plug the line . Suppose the natural gas is processed in a dehydration unit and the dew point is depressed 50° F . This means that no free water will exist in the gas until the temperature goes to 10° F or lower. Gas at 500 psia and 10° F contains about 5 Ibm of water vapor /MMCF. The dehydration unit must remove 25 Ibm of water from each 1 million cf of gas in order to achieve the 50° F dew point depression.

GAS DEHYDRATION TECHNIQUES

Techniques for dehydrating natural gas include:

1. Absorption using liquid desiccants.
2. Adsorption using solid desiccants.
3. Mechanical Refrigeration
4. Expansion Refrigeration.

Absorption Process

The more common liquids in use for dehydrating are diethylene glycol, triethylene glycol and tetraethylene glycol. Most glycol dehydrators use triethylene glycol, which can be heated to 340°F, Tetrathylene glycol is more expensive, but it can handle hotter gas without high losses and it can be heated up to 430°F.

Figure 16 shows how a typical bubble cap glycol contact tower works. Wet gas enters the base of the tower and flows upward through the bubble caps. Dry glycol enters the top of the tower, and because of the downcomer wier on the edge of each tray, flows across the tray and down to the next. There are typically six to eight trays in most applications. The bubble caps assure that the upward flowing gas is dispersed into small bubbles to maximize its contact area with the glycol.

Before entering the contactor the dry glycol is cooled by the outlet gas to minimize vapor losses when it enters the tower. The wet glycol leaves from the base of the tower and flows to the reconcentrator (reboiler) by way of heat exchangers, a gas separator, and filters, as shown in Figure 2

In the reboiler the glycol is heated to a sufficiently high temperature to drive off the water as steam. The dry glycol is then pumped back to the contact tower.

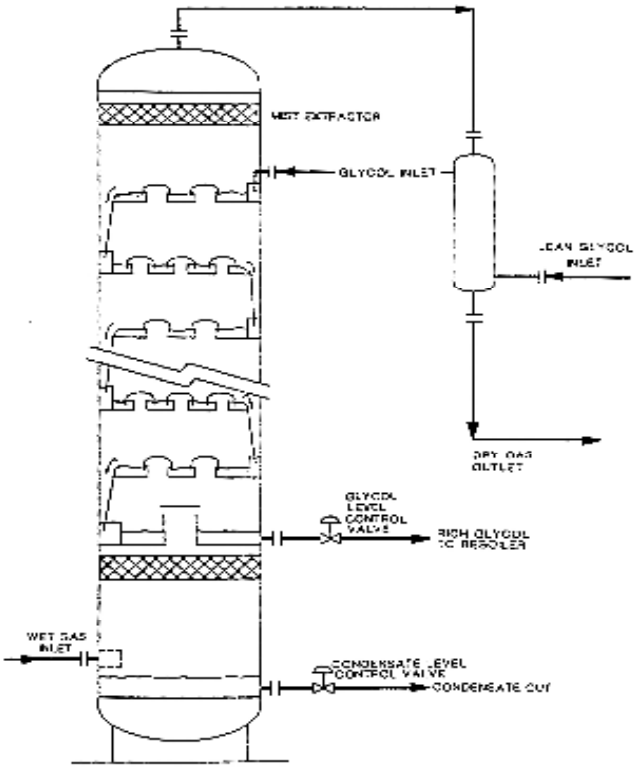


Figure 1. Typical glycol contact tower.

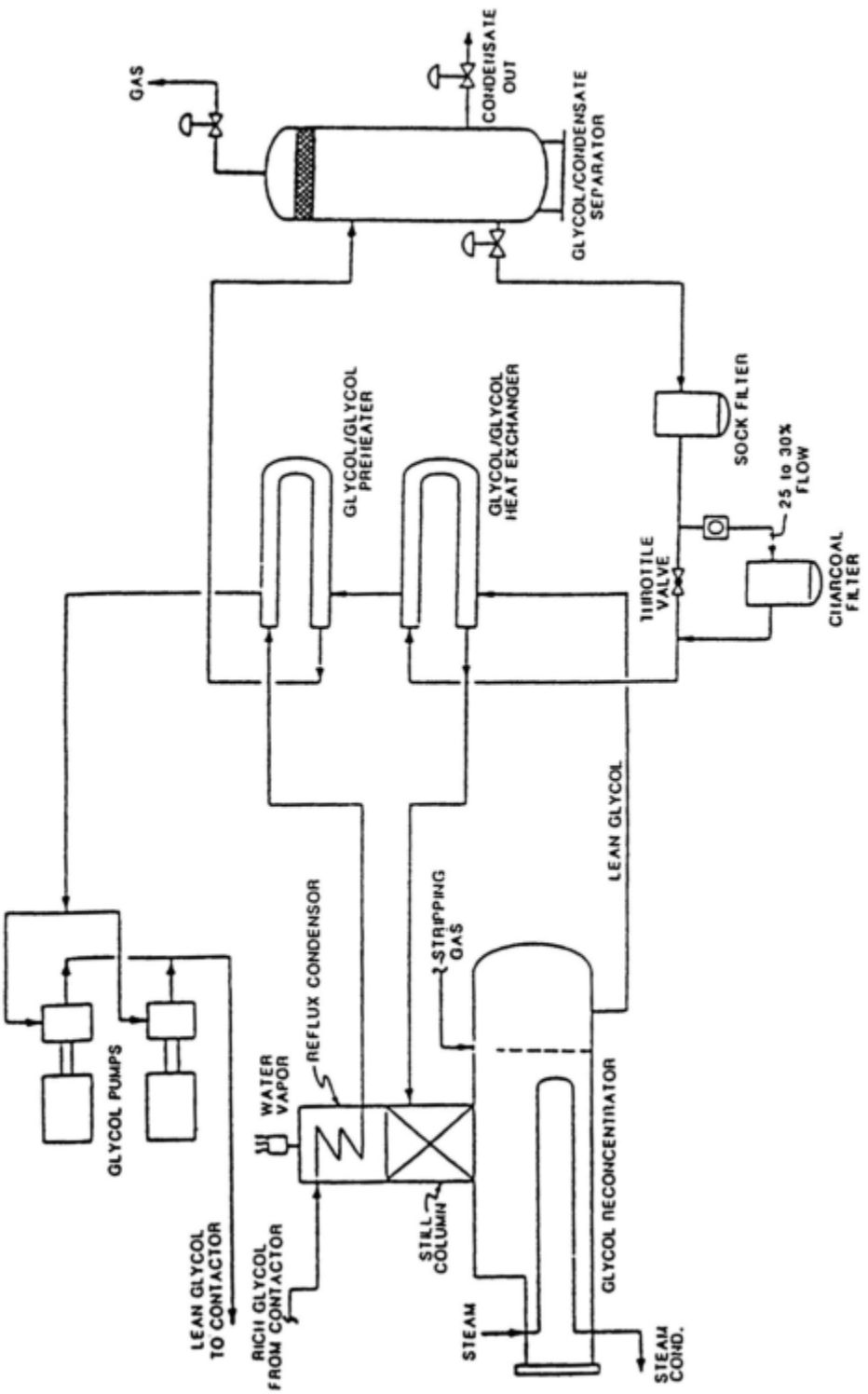


Figure 2. Typical glycol reconcentrator.

Adsorption Process

Solid bed dehydration systems work on the principle of adsorption. Adsorption involves a form of adhesion between the surface of the solid desiccant and the water vapor in the gas. The water forms an extremely thin film that is held to the desiccant surface by forces of attraction, but there is no chemical reaction. The desiccant is a solid, granulated drying or dehydrating medium with an extremely large effective surface area per unit weight because of a multitude of microscopic pores and capillary openings. A typical desiccant might have as much as 4 million square feet of surface area per pound.

Most solid desiccants used in natural gas dehydration are those adsorbents that can be "reactivated" or "regenerated" so that they may be used through many cycles of adsorption and reactivation. Some solid desiccants can produce exit water content as low as 1 ppmv or less. As a result they are widely used for feed streams to cryogenic processing systems.

The process is cyclic in nature, with a given bed being on adsorption for a given time and then heated to remove the adsorbed water. This means that for continuous operation a plant must contain at least two adsorbent beds. There must be a source of heat for heating the bed for regeneration and gas to flow through the bed to remove the desorbed water vapor. The temperature required for regeneration varies with the specific adsorbent being used.

Solid desiccants tend to adsorb heavy hydrocarbons and glycols either on the surface or in the pores. These adsorbates are difficult to remove during regeneration.

Multiple desiccant beds are used in cyclic operation to dry the gas on a continuous basis. The number and arrangement of the desiccant beds may vary from two towers, adsorbing alternately, to many towers. Three separate functions or cycles must alternately be performed in each dehydrator. They are an adsorbing or gas drying cycle, a heating or regeneration cycle, and a cooling cycle.

Figure 3 is a flow diagram for a typical two-tower solid desiccant dehydration unit. The essential components of any solid desiccant dehydration system are:

1. Inlet gas separator.
2. Two or more adsorption towers (contactors) filled with a solid desiccant.
3. A high-temperature heater to provide hot regeneration gas to reactivate the desiccant in the towers.
4. A regeneration gas cooler to condense water from the hot regeneration gas.
5. A regeneration gas separator to remove the condensed water from the regeneration gas.
6. Piping, manifolds, switching valves and controls to direct and control the flow of gases according to the process requirements.

In the drying cycle, the wet inlet gas first passes through an inlet separator where free liquids, entrained mist, and solid particles are removed. This is a very important part of the system because free liquids can damage or destroy the desiccant bed and solids may plug it. If the adsorption unit is downstream from an amine unit, glycol unit or compressors, a filter separator is preferred.

In the adsorption cycle, the wet inlet gas flows downward through the tower. The adsorbable components are adsorbed at rates dependent on their chemical nature, the size of their molecules, and the size of the pores. The water molecules are adsorbed first in the top layers of the desiccant bed. Dry hydrocarbon gases are adsorbed throughout the bed. As the upper layers of desiccant become saturated with water, the water in is heated to temperatures of 450°F to 600°F in the high-temperature heater and routed to the tower to remove the previously adsorbed water. As the temperature within the tower is increased, the water captured within the pores of the desiccant turns to steam and is absorbed by the natural gas. This gas leaves the top of the tower and is cooled by the regeneration gas cooler. When the gas is cooled the saturation level of water vapor is lowered significantly and water is condensed. The water is separated in the regeneration gas separator and the cool, saturated regeneration gas is recycled to be dehydrated. This can be done by operating the dehydration tower at a lower pressure than the tower being regenerated or by recompressing the regeneration gas.

Once the bed has been "dried" in this manner, it is necessary to flow cool gas through the tower to return it to normal operating temperatures (about 100°F to 120°F) before putting it back in service to dehydrate gas. The cooling gas could either be wet gas or gas that has already been dehydrated. If wet gas is used, it must be dehydrated after being used as cooling gas. A hot tower will not sufficiently dehydrate the gas.

The switching of the beds is controlled by a time controller that performs switching operations at specified times in the cycle. The length of the different phases can vary considerably. Longer cycle times will require larger beds, but will increase the bed life. A typical two-bed cycle might have an eight-hour adsorption period with six hours of heating and two hours of cooling for regeneration. Adsorption units with three beds typically have one bed being regenerated, one fresh bed adsorbing, and one bed in the middle of the drying cycle.

Internal or external insulation for the adsorbers may be used. The main purpose of internal insulation is to reduce the total regeneration gas requirements and costs. Internal insulation eliminates the need to heat and cool the steel walls of the adsorber vessel. Normally, a castable refractory lining is used for internal insulation.

The refractory must be applied and properly cured to prevent liner cracks. Liner cracks will permit some of the wet gas to bypass the desiccant bed. Only a small amount of wet, bypassed gas is needed to cause freezeups in cryogenic plants.

Ledges installed every few feet along the vessel wall can help eliminate this problem. The initial cost for a solid bed dehydration unit generally exceeds that of a glycol unit. However, the dry bed has the advantage of producing very low dew points, which are required for cryogenic gas plants, and is adaptable to very large changes in flow rates. Disadvantages are that it is a batch process, there is a relatively high pressure drop through the system, and the desiccants are sensitive to poisoning with liquids or other impurities in the gas.

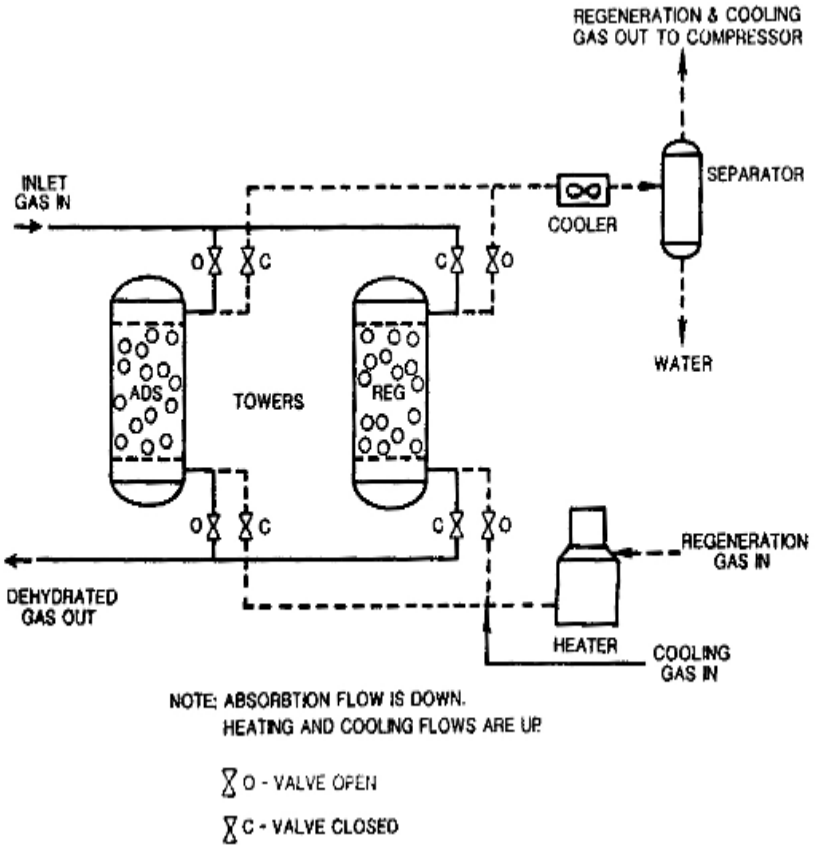
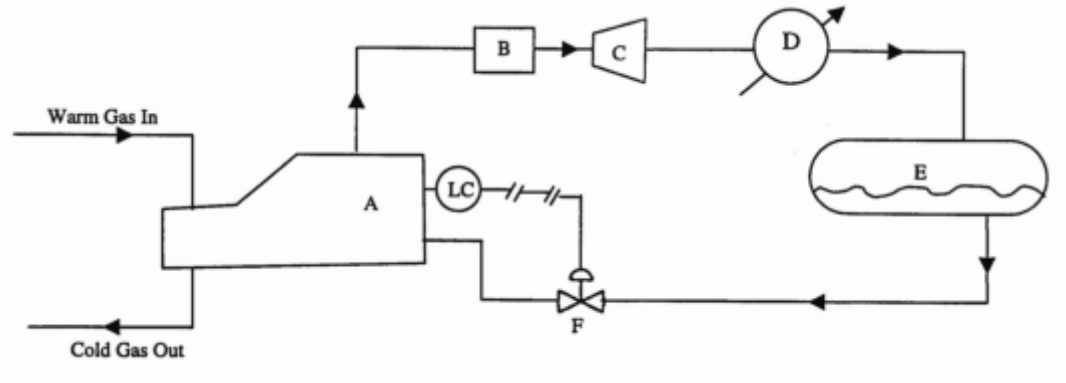


Figure 3. Simplified flow diagram of a solid bed dehydrator.

Mechanical Refrigeration Process

Liquid propane is vapourized in a process chiller (A), by removing heat from a process stream. The resultant cold propane vapour passes through a knockout pot (B) to remove any liquid, which could damage the compressor. The compressor (C) raises the propane vapour to a high enough pressure that it can be completely condensed to a liquid in a water or air cooled condenser (D). Liquid propane from the surge drum (E) passes through a throttling valve (F) to maintain a level in the chiller shell. The cycle. then repeats.

The cycle can be thought of as heat pumping. Heat is brought into the circuit at a low temperature level and "pumped up" so that it can be discarded through the condenser to a high temperature heat sink. Propane acts as the heat carrier. Obviously the amount of power used in the cycle will depend on how low the chiller temperature is and how high the condensing temperature is (the "head" of the heat pump), as well as how much heat is being removed (the "flow volume" of the heat pump). The chiller temperature will have to be lower than process stream exit temperature, and the condensing temperature will have to be higher than the heat sink temperature. The closeness of approach designed into these heat exchangers will be a judgments decision, taking into account the cost of the heat exchangers on the one hand and both investments and power cost for the compressor on the other however, this process is only applicable when high water contents are acceptable (+30 lb/MMscf)



A	Chiller	D	Condenser
B	Knokout pot	E	Surge Drum
C	Compressor	F	Throttling Valve

Figure 4 Propane refrigeration

Expansion Refrigeration Process

Expansion refrigeration is a process during which gas is cooled as a result of reducing its pressure. Pressure reduction is caused by a choke at the wellhead or upstream of the refrigeration unit, which is also called low temperature separation unit (LTS unit).

LTS units are designed to allow hydrates to form and to melt them with the heat of the incoming gas stream before they can plug downstream equipment.

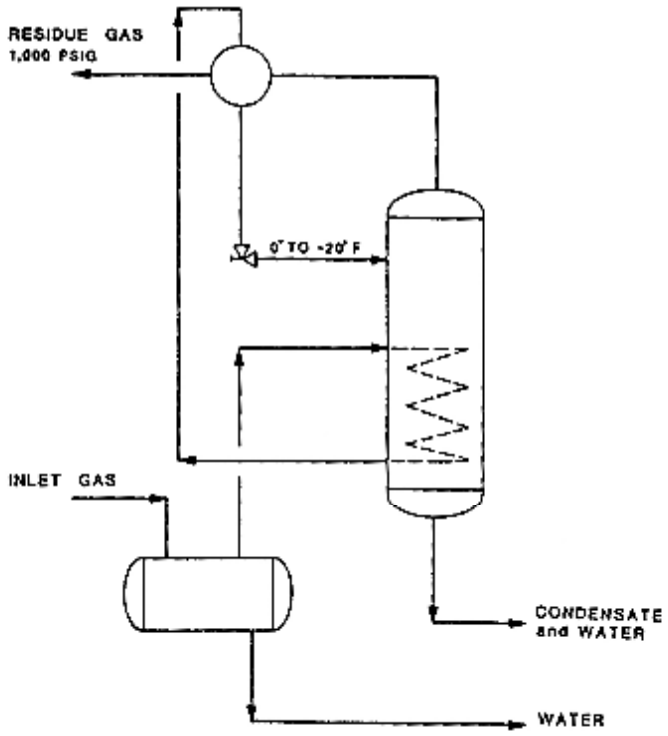


Figure 5 simplified flow diagram of a typical LTS unit

The inlet gas stream is choked at the well to 2,000 to 3,000 psi or until the temperature declines to approximately 120˚F. The condensate and water are separated in a high pressure separator. The free water otherwise would freeze in the choke of the low-temperature separator as the expanding gas drops the temperature below the hydrate point. Liquid hydrocarbons passing the choke do not contribute to the adiabatic expansion and only give up heat, thus decreasing the net refrigeration effect of the expanding gas.

The hot gas stream enters a coil in the bottom of the low-temperature separator at a temperature of 120° or greater. The inlet gas is cooled to just above the hydrate formation temperature with the outlet coming off the low temperature separator to assure the lowest possible temperature when it enters the vessel after the choke. This choke is mounted in the vessel itself. When the pressure drop is taken, the temperature will decrease to well below the hydrate point. Hydrates form, but they fall into the bottom of the separator and are melted by the heating coil. The hydrates do not plug the choke because the choke is inside the separator.

The gas, condensate, and free water are then discharged from the vessel through backpressure and liquid dump valves. The gas leaving the separator is saturated with water vapor at the temperature and pressure of the low temperature separator. If this temperature is low enough, the gas may be sufficiently dehydrated to meet sales specifications.

In summary, a colder separation temperature removes more liquid - from the gas stream; adequate bottom heating melts the hydrates and revaporizes the lighter components so they may rejoin the sales gas instead of remaining in liquid form to be flashed off at lower pressure; and cold refluxing recondenses the heavy components that may also have been vaporized in the warming process and prevents their loss to the gas stream.

LTX units are not as popular as they once were. The process is difficult to control, as it is dependent on the well flowing-tubing pressure and flowing-tubing temperature. If it is being used to increase liquid recovery, as the flowing temperature and pressure change with time, controls have to be reset to assure that the inlet is cold enough and the coil hot enough. If the coil is not hot enough, it is possible to destabilize the condensate by increasing the fraction of light components in the liquid stream. This will lower the partial pressure of the intermediate components in the stock tank and more of them will flash to vapor. If the inlet stream is not cold enough, more of the intermediate components will be lost to the gas stream.

From a hydrate melting standpoint it is possible in the winter time to have too cold a liquid temperature and thus plug the liquid outlet of the low temperature separator. It is easier for field personnel to understand and operate a line heater for hydrate control and a multistage flash or condensate stabilizer system to maximize liquids recovery.

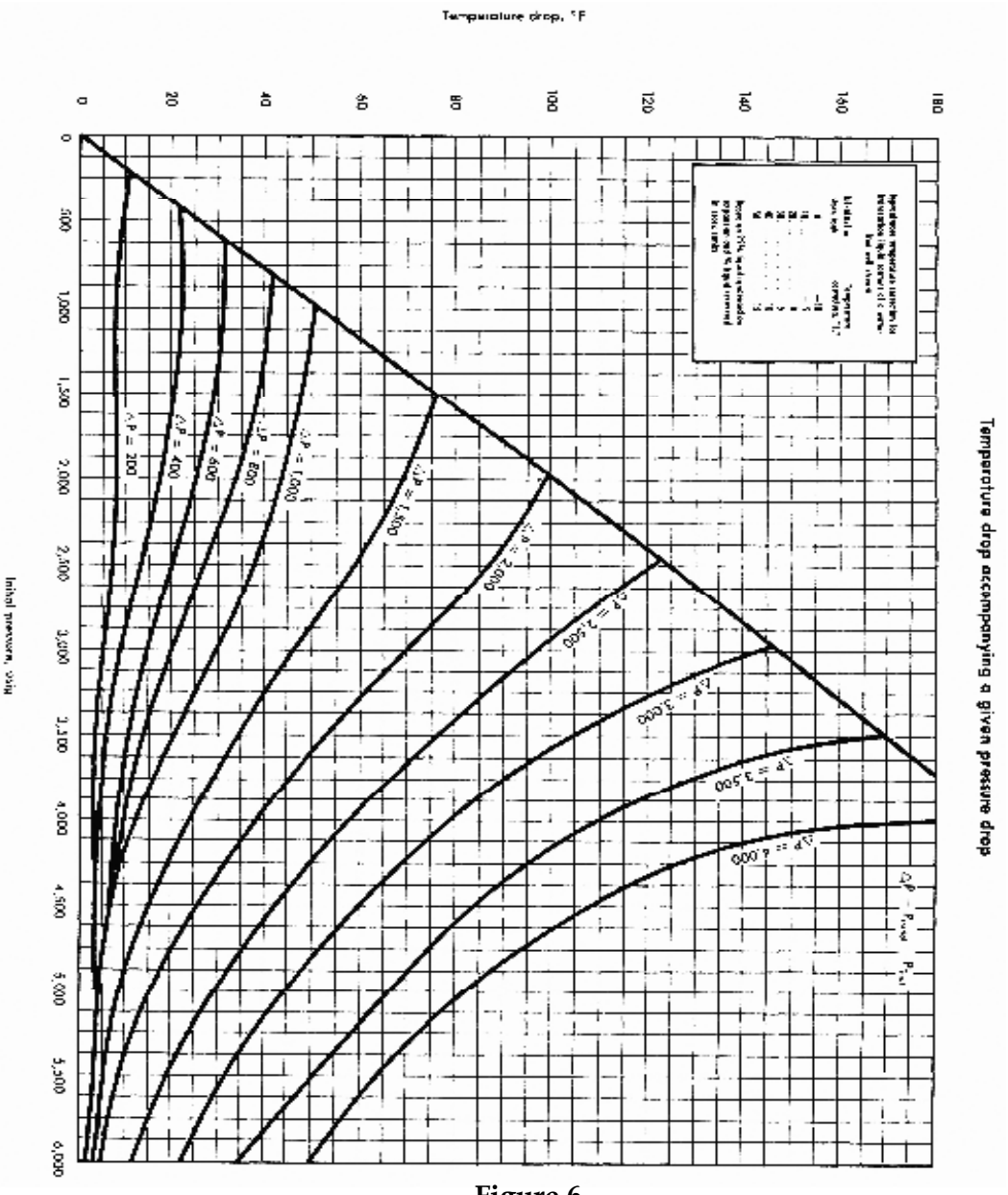


Figure 6

GLYCOL DEHYDRATION AND REGENERATION SYSTEMS

INTRODUCTION

Most natural gas streams leaving the reservoir contain water vapor. In many cases free water is also produced along with natural gas.

Natural gas cools as it travels up the wellbore to the surface due to pressure reduction and conduction of heat through the pipe to cooler formation walls. Since the ability of gas to hold water vapor decreases as gas temperature decreases, produced natural gas is nearly always saturated with water vapor when it reaches the surface.

Additional cooling of saturated gas will cause free water to form. If the gas is further cooled, hydrates will form and serious equipment damage or flow restrictions will result. This is why it is so important to remove water vapor from natural gas.

DEFINITIONS

Water in Natural Gas

All unprocessed natural gas contains water, either in liquid or vapor form. The presence of water in natural gas causes two major problems in transmission lines:

- Corrosion which causes pitting and pipeline damage.
- Hydrates which restrict the flow of gas.

Dehydration

The process of removing water from a substance is called dehydration. Although there are several methods for removing water from gas, the most commonly used dehydration method utilizes a substance known as Triethylene Glycol (TEG) or simply glycol.

Glycol

Triethylene Glycol, TEG, and glycol all refer to the same substance. TEG is expensive but it is efficient to remove the water, recycling the TEG to be used over and over again.

Absorption

Much like a sponge, glycol is used to absorb water from natural gas. By mixing together the wet gas and glycol, water is absorbed by the glycol, thereby removing it from the natural gas.

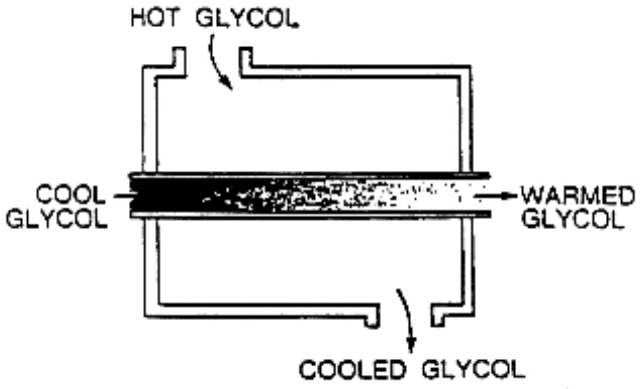
Distillation

In distillation, water is separated and removed from glycol by boiling. Glycol does not begin' to boil until approximately 435° F (224° C).
Water boils at 212° F (100° C). Distillation of water from glycol involves heating the glycol-water mixture to a temperature between 212° F (100° C) and 400° F (204°C) allowing water to separate as vapor.

Heat Transfer

Condensation is the process by which heat travels through a substance. If two containers of water - one containing cold water, the other hot water - are brought into contact with each other, the temperature of the hot water container will decrease and the temperature of the cold water container will increase. The temperature changes result because of heat transfer from hot to cold.

FIGURE - 1
(CONDUCTION)



By bringing together "cool" and "hot" glycol in heat exchangers, the process of heat transfer through condensation is accomplished allowing for temperature control of the dehydration process. In glycol dehydration, it is important to maintain fluid temperatures within relatively narrow ranges, to optimize the efficiency of the process. Improper temperature control can cause glycol foaming.

Liquid Desiccants

A number of liquids possess the ability to absorb water from gas. Yet, there are very few which meet the criteria for a suitable commercial process: are highly **hygroscopic,** do not solidify in a concentrated solution, non-corrosive, do not form precipitates with gas constituents, easily regenerated to a high concentration, can be separated easily, essentially nonsoluble in liquid hydrocarbons, and relatively stable in the presence of sulfur compounds and carbon dioxide under normal operating conditions.

A liquid desiccant is any liquid which removes water from another substance when the two come in contact. Common liquid desiccants used in the oil and gas industry to dehydrate natural gas are methanol, ethylene glycol, diethylene glycol, triethylene glycol and tetraethylene glycol. Methanol, ethylene glycol, and diethylene glycol are commonly used in injection systems as hydrate inhibitors.

DEG (Diethylene) is somewhat cheaper to buy and sometimes is used for this reason. But, by the time it is handled and added to the units there is no real saving. Compared to TEG, DEG has a larger carry-over loss, offers less dewpoint depression and regeneration to high concentrations is more difficult. For these reasons, it is difficult to justify a DEG unit, although a few units are built each year.

TEG (Triethylene) is preferred for use in dehydration units because:

- It is more easily regenerated due to its high boiling point and other physical properties.
- It has a high decomposition temperature of 404°F (207°C)
- It has lower vaporisation losses than other glycols
- It has lower capital and operating costs than other glycol systems

TREG (Tetraethylen) is more viscous and more expensive than the other processes. The only real advantage is its lower vapor pressure which reduces absorber carry-over loss. It may be used in those relatively rare cases where glycol dehydration will be employed on a gas whose temperature exceeds about 50°C [122°F].

TABLE 1 PHYSICAL PROPERTIES OF HYDRATE INHIBITORS

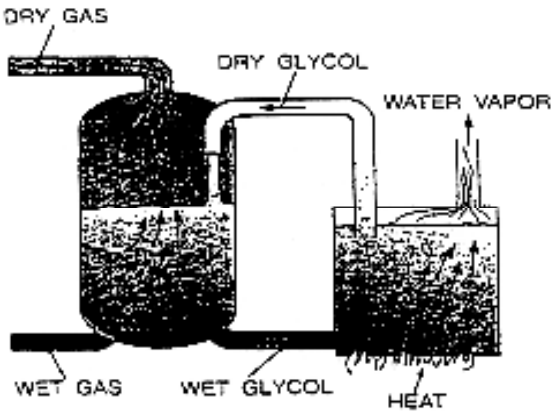
	Ethylene Glycol	Diethylene Glycol	Triethylene Glycol	Tetraethylene Glycol	Methanol
	HOC_2H_4OH	$HO(C_2H_4O)_2H$	$HO(C_2H_4O)_3H$	$HO(C_2H_4O)_4H$	CH_3OH
Molecular Weight	62.07	106.12	150.17	194.23	32.04
Apparent Specific Gravity at 20/20°C.	1.1155	1.1184	1.1255	1.1757	0.7924
Δ Sp. Gr./Δ t. at 10 to 40°C.	0.00070 per °C.	0.00072 per °C.	0.00070 per °C.	0.00080 per °C.	0.00094 per °C.
Boiling Point at 760 mm. Hg	197.6°C.	245.8°C.	288°C.	Decomposes	64.5°C.
at 300 mm. Hg	108°C.	213°C.	252°C.	291	43°C.
at 10 mm. Hg	91°C.	129°C.	161°C.	190	-16°C.
Δ B.p./Δ p. at 750 to 770 mm. Hg	0.045°C. per mm.	0.050°C. per mm.	0.055°C. per mm.	--	0.033 C per mm.
Vapor Pressure at 20°C.	<0.1 mm. Hg	<0.01 mm. Hg	<0.01 mm. Hg	<0.01 mm. Hg	97 mm. Hg
Surface Tension at 25°C.	47.0 dynes per cm.	44.8 dynes per cm.	45.2 dynes per cm. at 20°C.	--	22.0 dynes per cm.
Viscosity at 0°C.	57 cps.	121 cps.	172 cps.	270 cps.[c]	0.9 cps.
at 20°C.	21 cps.	36 cps.	49 cps.	55 cps.	0.6 cps.
at 40°C.	9.5 cps.	15 cps.	19 cps.	20 cps.[c]	0.5 cps.
Freezing Point	-12.7°C.	-7.8°C.	-4.3°C.	-4.0°C.	-97.7°C.
Refractive Index at 20°C., n_D	1.4319	1.4474	1.4561	1.4598	1.3286
Δ n_D/Δ t. at 20 to 40°C.	0.00028 per °C.	0.00030 per °C.	0.00030 per °C.	0.00030 per °C.	0.00040 per °C.
Water Solubility at 20°C.	Complete	Complete	Complete	Complete	Complete
Solubility in Organic Solvents at 25°C.					
Acetone	8% by wt.	Complete	Complete	Complete	Complete
Benzene	<0.1% by wt.	3.4% by wt.	Complete	Complete	Complete
Carbon Tetrachloride	1.4% by wt.	0.28% by wt.	0.4% by wt.	3.6% by wt.	Complete
Ethyl Ether	3.1% by wt.	6.3% by wt.	6% by wt.	4.8% by wt.	Complete
Heptane	<0.2% by wt.	0.02% by wt.	<0.03% by wt.	<0.07% by wt.	2% by wt.
Methanol	Complete	Complete	Complete	Complete	Complete
Heat of Vaporization at 760 mm. Hg	364 Btu per lb.	232 Btu per lb.	174 Btu per lb.	--	474 Btu per lb.
at 300 mm. Hg	381 Btu per lb.	240 Btu per lb.	185 Btu per lb.	--	
Heat of Combustion at 25°C.	8780 Btu per lb.	9552 Btu per lb.	10,020 Btu per lb.	10,310 Btu per lb.[c]	9751 Btu per lb.
Critical Constants (estimated values)					
Temperature	372°C.[c]	408°C.[c]	--	--	240°C.
Pressure	76 atm.[c]	46 atm.[c]	--	--	78.5 atm.
Volume	0.186 liters per mole[c]	0.316 liters per mole[c]	--	--	0.118 liters per mole
Specific Heat, Cal./g./°C.[a] $C_p = A + BT + CT^2$					
A	0.53806	0.50330	0.49487	0.4603	0.57112
B × 10^3	11.21	8.40	7.31	10.00	11.16
C × 10^6	0.006	5.59	4.89	2.10	8.05
Flash Point	240°F.[c]	290°F.[c]	330°F.[c]	360°F.[c]	58°F.

Basic Processes

There are two basic purposes of a glycol dehydration unit.

- To dry natural gas before puffing it into a pipeline.
- To remove water from glycol so that it can be used over and over again in the dehydration process.

FIGURE 2
BASIC PROCESS



The process, while it may seem somewhat complicated, is actually quite simple. Natural gas and glycol flow counter-current in a contractor tower. Glycol literally soaks up the water, leaving dry gas. The wet glycol then goes through a process of distillation where the water is removed by boiling. The dry glycol is then sent back to function again in the dehydration of gas. In this manner, glycol is recycled.

ABSORPTION USING LIQUID DESICCANTS (GLYCOL DEHYDRATION)

The glycol ability to absorb water is one of its characteristics that have led to its widespread use in the dehydration of natural gas. The use of glycol to dehydrate natural gas began in the 1930's and its use has steadily increased through the years.

The first glycol plants obtained dew point depressions in the range of 20-50° F. Today after many years of experience and the increasing technical development, it is possible to design glycol dehydrations which will obtain dew point depressions of over 100° F. The early plants used diethylene glycol and triethylene glycol came into use in 1949.

Most process units (Figure 3) using absorption for dehydration employ triethylene glycol (TEG) as the absorbent. Other glycols and glycol mixtures are used, but the relative number of such units is small and their design details are similar to a TEG plant.

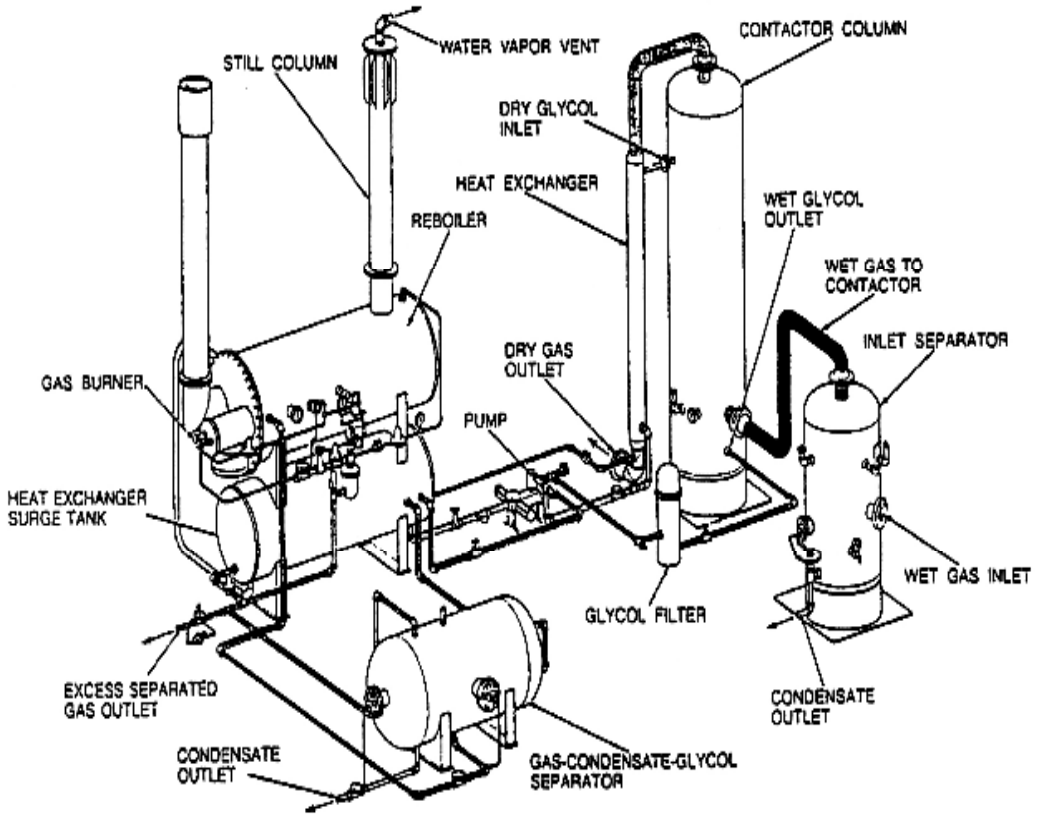


FIGURE 3 GLYCOL DEHYDRATION UNIT

System Components

The efficient operation of a glycol dehydration system requires good understanding of the dehydration process and process variables as well as frequent monitoring of dew points. Better dehydration results, as well as significant savings, are experienced when systems are operated properly. The operation of individual components within the system must be understood to properly operate and troubleshoot the system.

A typical glycol dehydration system is shown in Figure 4. It consists of the following components: contactor column, reboiler, glycol filter, pump, surge tank, gas-condensate glycol separator and heat exchangers.

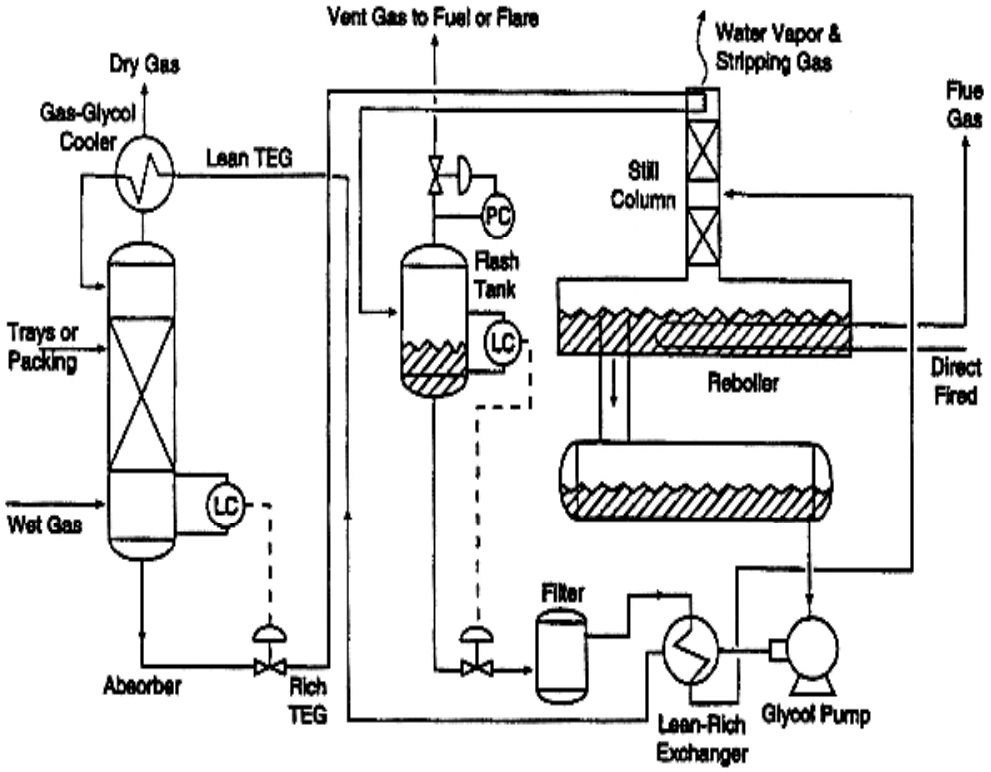


Figure 4 Basic Glycol Dehydration unit

Contactor Column

The function of the contactor column (Figure 5) sometimes called an absorber, is to contact natural gas with the glycol, so that the glycol can remove water vapor from the natural gas. These vessels are designed to accommodate a certain gas volume and pressure. It should be noted that to exceed design specifications will increase glycol losses and outlet gas dew point.

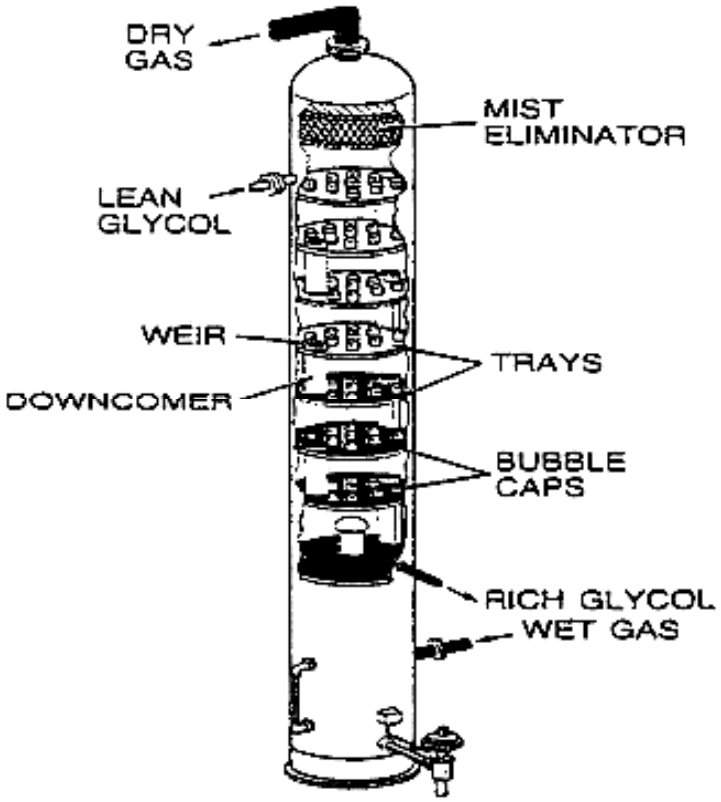


FIGURE 5 TRAY CONTACTOR COLUMN

For large volumes of gas, the contactor is usually a tray column containing 4 to 12 trays on which up flowing gas bubbles through down flowing glycol, The number of trays in the contactor will affect the amount of moisture removed from the gas by the glycol; more trays mean more moisture removal.

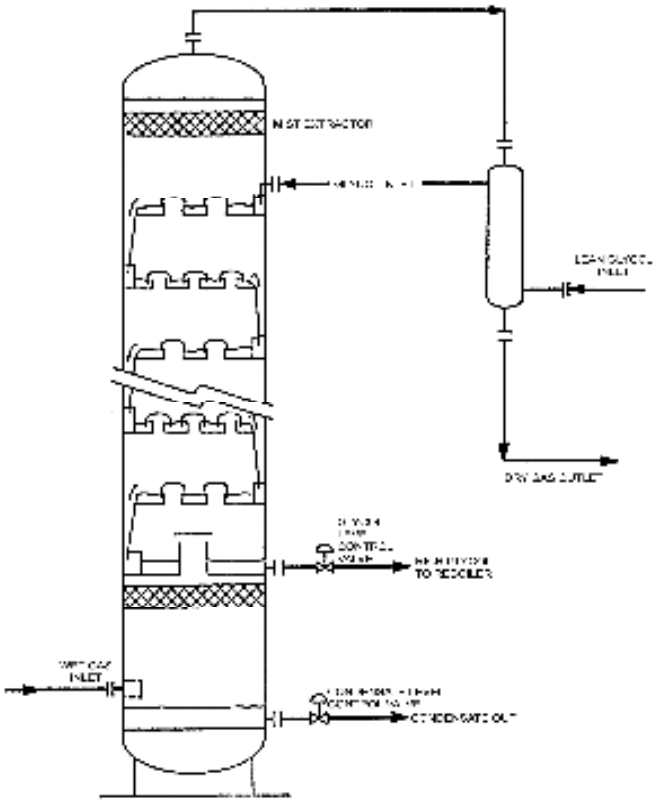


Figure 6 - Schematic flow diagram of a typical glycol contact tower.

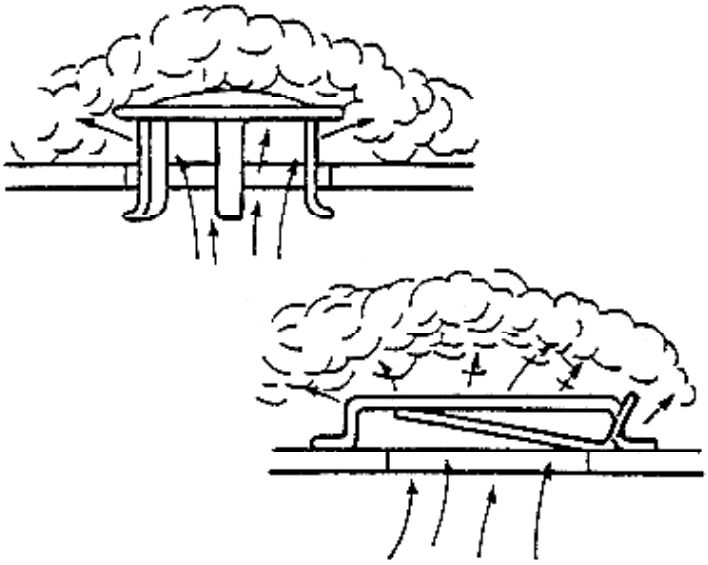


Figure 7 – Vapor Flow Through Valves

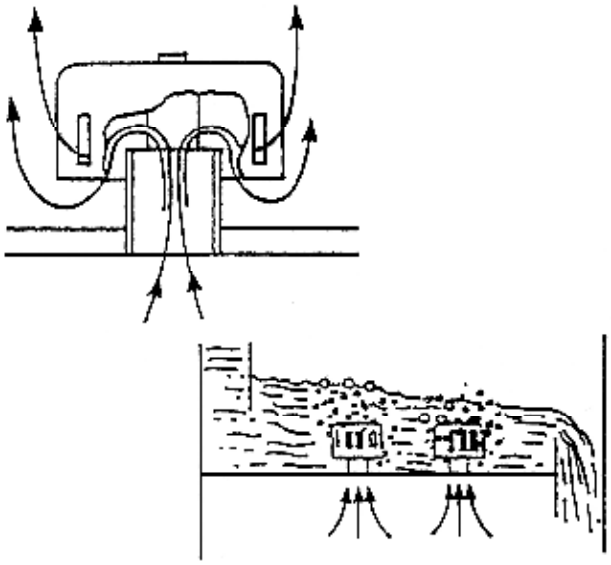


Figure 8 – Vapor Flow Through Bubble Caps

Rarely does the number of trays exceed ten. Each has a number of openings with bubble caps bolted over them to disperse gas through the glycol solution as the gas is forced to pass through these caps and bubble evenly through the glycol. The gas gives up water and becomes drier as it passes upward through each succeeding tray. The glycol becomes more saturated with water as it flows downward over each tray. Also found on the trays are weirs and downcomers. Weirs, which are dam-like devices, are used to maintain the level of glycol above slots in the bubble caps. Downcomers carry the glycol to the trays below.

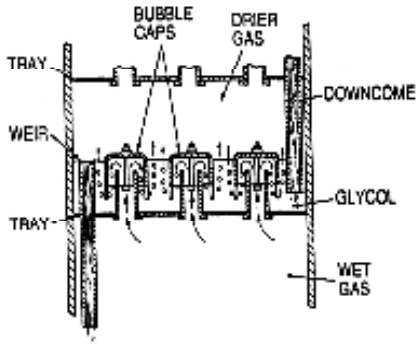


FIGURE 9
TRAY WITH BUBBLE CAPS

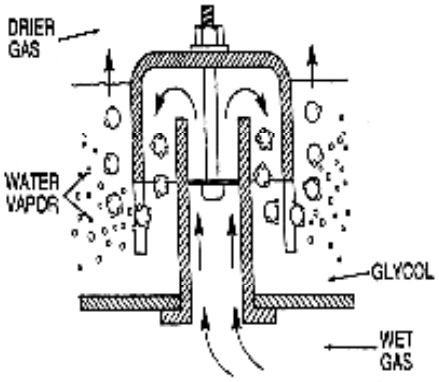


FIGURE 10
BUBBLE CAP

In smaller capacity units, that is, contactors having a diameter of 18 inches or less, random packing may be used instead of trays. The packing is metal, plastic, or ceramic structures that are designed to furnish a large surface area for the glycol solution to spread out and make better contact with the gas. Random packing is poured into the contactor onto a support gad.

Four feet packing is usually standard and sufficient for dew point depressions up to 55°F to 65°F (13°C to 18°C). If higher dew point depressions are required, additional packing may be required.

Packed columns (Figure 11) utilise the same process as tray columns, that is, glycol flows down over the packing and gas flows up through the packing contacting the glycol. Packed columns are less expensive; however, the glycol tends to channel easier and have poorer flow distribution. Therefore, special attention must be given to the design of a glycol distribution header above the packing so that gas/glycol contacting will be continuous throughout the packing and the glycol will not channel.

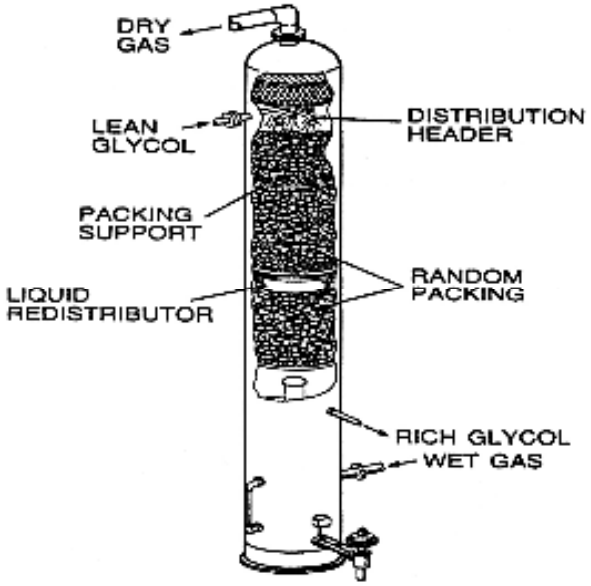


FIGURE 11
Packed Contactor Column

Contactors being designed today may contain structured packing (Figure 12a) Structured packing is a group of corrugated metal sheets welded into a specific pattern and placed in the contactor on edge. Glycol coats these sheets and the gas flows between them. This type of packing is much more efficient than bubble caps or random packing. Structured packing is used in columns from six inches (15 centimeters) in diameter up to ten feet (three meters).

FIGURE 12 a
a STRUCTURED PACKING

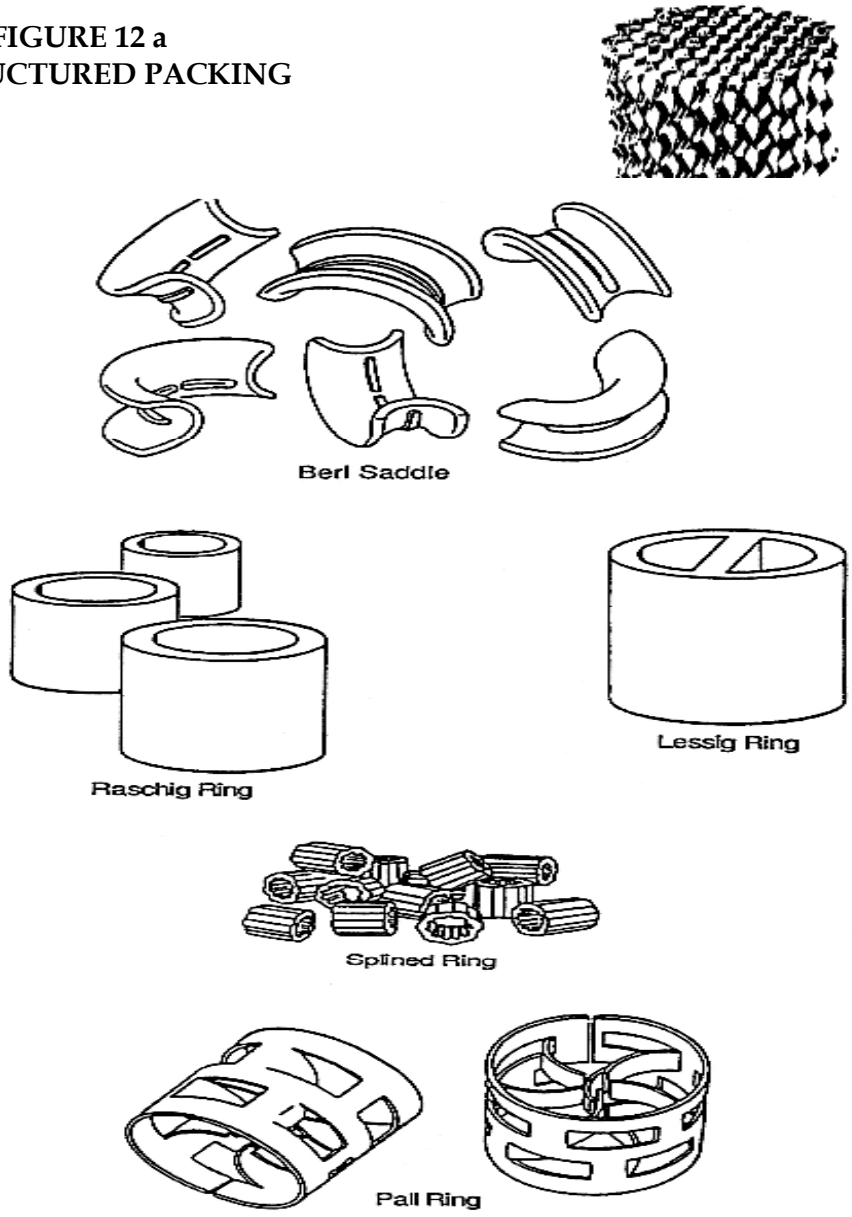


Figure 12 b – Various types of packing

Before lean glycol enters the contactor, two things happen:

- The glycol is pumped up to contactor pressure
- The glycol temperature is lowered to 10-15°F (5-7°C) above the inlet gas temperature. This is done by passing the lean glycol through a heat exchanger where it is cooled by the dry gas leaving the contactor

It is best to have the glycol entering the contactor 10-15°F (5-7°C) warmer than the inlet gas. Cooler glycol causes condensing of hydrocarbon vapors which cause foaming, resulting in greater glycol loss. Warmer glycol reduces effectiveness of absorbing water from the gas. The glycol/gas heat exchanger is sized to give the correct glycol temperature and usually does not require attention. Incorrect glycol temperatures entering the contactor indicate an imbalance in the glycol to gas flow rate, a problem in the heat exchanger, or the reboiler temperature is out of adjustment.

Above the top tray there is usually space for separation where most of the entrained glycol particles in the gas stream will settle out. Glycol not settling out will be removed by a mist eliminator (Figure 13) at the top of the contactor. This prevents liquid glycol from being carried out of the contactor and into the gas discharge line, commonly referred to as carryover.

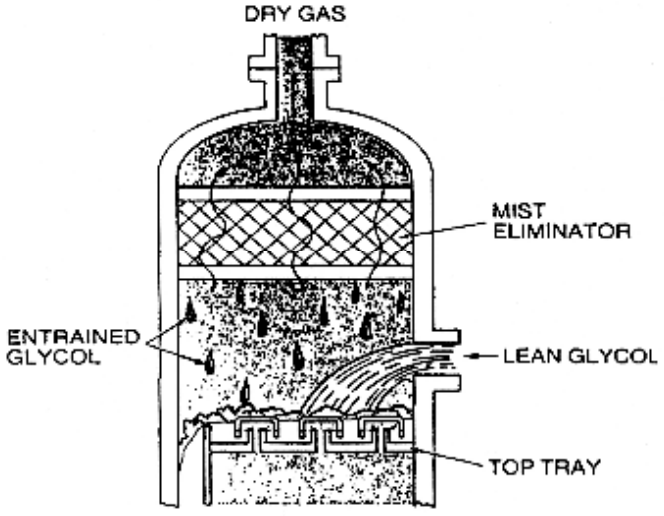


Figure 13 Mist Eliminator

Carryover (Figure 14) can result from foam build-up caused by glycol contamination; however, carryover can also be caused from a high gas rate. It will happen continuously when the gas rate is high enough to agitate the liquid at the top tray so that a foam forms that is too thick for the mist eliminator to handle. When this happens, the gas rate must be reduced to eliminate carryover.

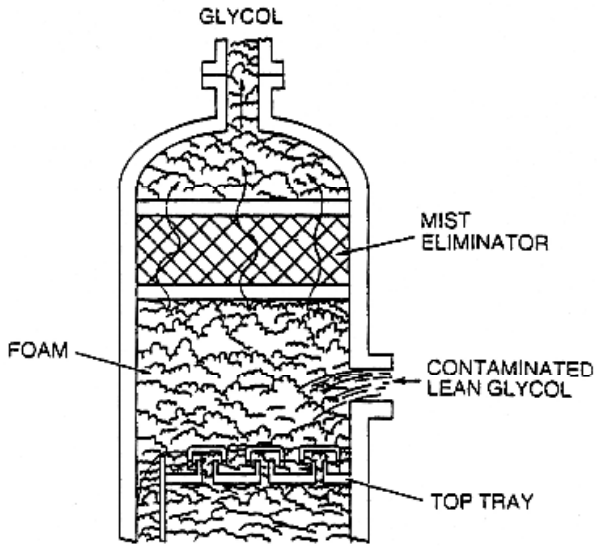


FIGURE 14 GLYCOL CARRYOVER

Level control on the contactor is important in stabilising operation. The level controller should be adjusted to hold a uniform flow rate of glycol out of the contactor. It is more important to hold a constant flow of liquid and let the level in the contactor vary a few inches than to hold a constant level and let the flow fluctuate. Flow rate surges will cause the reboiler to operate inefficiently and may overload it.

Gas-Condensate-Glycol Separator

separator is also known as the flash tank or Glycol-gas separator (Figure 15). It is used to recover gas which dissolved in the glycol solution in the contactor as well as any liquid hydrocarbons (condensate) carried out of the contactor by the glycol solution.

The G-C-G separator is simply a three phase separator in which gas, liquid hydrocarbons, and glycol are separated. The gas leaves the top of the vessel and is vented or may be used to supplement the fuel gas required for the reboiler. A pressure control valve maintains a back pressure on the vessel, usually 50 to 85 psig (345 to 587 kPa). Any excess gas is discharged through this back pressure valve.

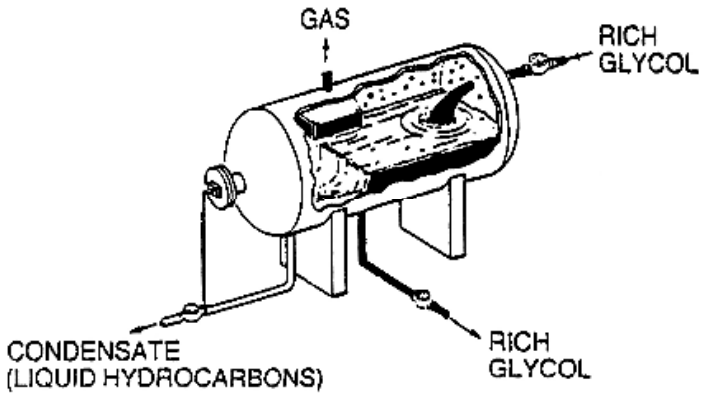


FIGURE 15 GAS-CONDENSATE-GLYCOL SEPARATOR

Separating liquid hydrocarbons from glycol before they enter the reboiler reduces the load on the carbon filter and helps prevent carbon from building up on the reboiler firetube. Problems caused by hydrocarbons entering the still column are flooding, glycol loss and possible damage to the still column.

Two level control systems are typically installed on gas-condensate-glycol separators. The upper system regulates flow of liquid hydrocarbon discharged from the vessel. The lower system regulates the flow of glycol from the vessel.

Filters

Filters are installed in the glycol stream to remove solids and other contaminants which may cause plugging and foaming. There are two types of filters commonly used in gas dehydration systems: One type is used for solids removal and the other for dissolved contaminants removal.

Solids Removal: Fine screen, sock type or cartridge filters are used for solids removal. Solid particles can cause erosion of pump pistons, valve seals and discs, plugging of equipment, and foaming.

Sock filters are the most commonly used type for solids removal. Sock filters contain cylindrical elements that are replaced as they become coated. A pressure drop of 3-6 psig (21-41 kPa) usually occurs as glycol flows through the elements; however, the pressure drop increases as the elements remove solid material and become plugged.

To measure the pressure drop across elements, a differential pressure gauge is installed. When the pressure drop rises to 15-20 psig (104-138 kPa), the elements should be replaced so that collapse of the elements or stoppage of glycol does not occur.

Dissolved Contaminants Removal: Activated carbon filters are recommended for the removal of dissolved contaminants. They work well until their adsorption capacity is reached. In cases where the glycol contains appreciable quantities of light hydrocarbons, they must be changed or reactivated frequently. Most glycol systems cannot be operated successfully without carbon adsorption.

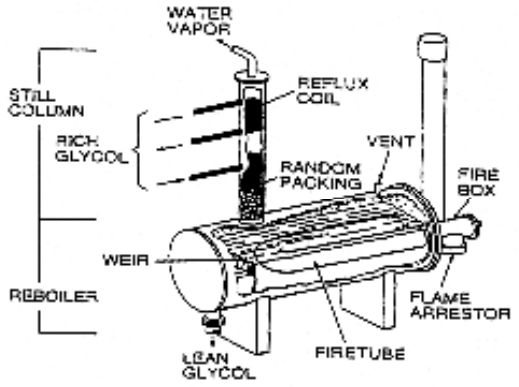
There are two types of activated carbon filters. Most systems use a carbon canister; however, larger systems use a loose fill carbon vessel. When the loose fill vessel is used, care must be taken to trap carbon fines and keep them from entering the glycol dehydration system. This is especially important when a fresh carbon bed is put into service. Routing the glycol flow through the carbon bed prior to the particle filter is the usual method of trapping carbon fines. However, care must be taken to reroute the glycol flow to its proper order or the life of the carbon bed will be shortened.

Carbon filters should be replaced anytime the level of contaminants in the glycol solution goes up. Monthly test should be run to determine the contamination level.

Regenerator and Still Column

The regenerator (Figure 16) is a combination of the glycol reboiler and the still column. They operate together to regenerate the rich glycol, making it lean again and ready for use in the contactor column.

FIGURE 16
REGENERATOR



The reboiler is the vessel which supplies heat to separate glycol and water by simple distillation. Glycol is heated to a temperature between 380°F and 400°F (193°C and 204°C) to remove enough water vapor to regenerate the glycol to 98.5-99%. The temperature of triethylene glycol should not exceed 400°F (204°C) because TEG will begin to break down at higher temperatures.

Reboilers can be direct fired or heated by steam or hot oil. The direct fired type, containing a removable fire box and firetube, is normally used.

The heating element should be conservatively sized to insure long tube life and prevent glycol decomposition by overheating the glycol.

Glycol level in the reboiler is maintained by an overflow weir. Excess glycol spills over the weir and flows downward into the surge tank by gravity. When glycol level is low in the surge tank, fresh glycol is added to the reboiler, so that it can be dried before going to the surge tank.

The still column, sometimes called a stripper, is the vessel located on top of the reboiler where distillation of glycol and water actually takes place. Still columns are normally packed columns and have finned condensers or reflux coils in the top to cool water vapor leaving the column and to recover entrained glycol.

The amount of glycol lost with the water vapor leaving the still column is controlled by the temperature of the water vapor, normally referred to as still column overhead temperature. This temperature should be set at the boiling point of pure water for the pressure in the top of the still column.

Normally the pressure will be atmospheric, 14.7 psia at sea level, and the temperature will be 212°F (100°C). Systems with back pressure will require higher temperatures. Systems operating above sea level will require lower temperatures due to the lower pressures.

If the still overhead temperature is above the boiling point of pure water, glycol carryover will occur and losses will be higher than normal. If the still overhead temperature is below the boiling point of pure water, too much water will be condensed and the reboiler heat requirements and fuel usage will increase.

Column flooding will result when glycol being fed to the still column has a high concentration of light hydrocarbons, which will distil out of the glycol and condense in the upper portion of the column. Flooding will also result if the glycol flow rate exceeds maximum design limits. Column flooding does not allow proper glycol/water separation and rich glycol can be lost through the vent line.

Surge Tank

This is the vessel (Figure 17) used to store regenerated glycol for pump suction. It is commonly positioned under the reboiler in the glycol dehydration system. The surge tank should be vented and the vent line kept unplugged. Vapors, which are trapped in the surge tank, could cause the circulation pump to vapor lock. The vent line should be piped away from process equipment.

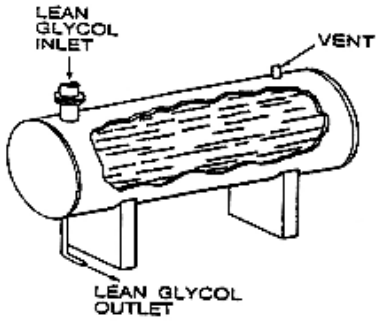


FIGURE 17
SURGE TANK

Pumps

Most small dehydration units use a fluid driven pump while larger units generally use an electrically driven reciprocating pump. In either case, the pump is a critical part of the unit because it has the only moving parts in the system. Before entering the pump, rich glycol passes through a strainer to remove large particles.
Leaks can be a problem around glycol pumps. Proper packing gland maintenance and the use of manufacturer's recommended packing can minimize this problem.

Heat Exchangers

There are two types of heat exchangers used in a glycol dehydration unit: glycol to gas heat exchanger and glycol to glycol heat exchanger(s).

Glycol/Gas Heat Exchanger. Dry gas leaving the contactor passes through this heat exchanger, where the temperature of the gas is raised slightly as it cools the incoming lean glycol. Thus, the final stage of lean glycol cooling is accomplished here. Incoming glycol is cooled to a temperature about 10-15°F (5-7°C) above that of the natural gas entering the contactor.

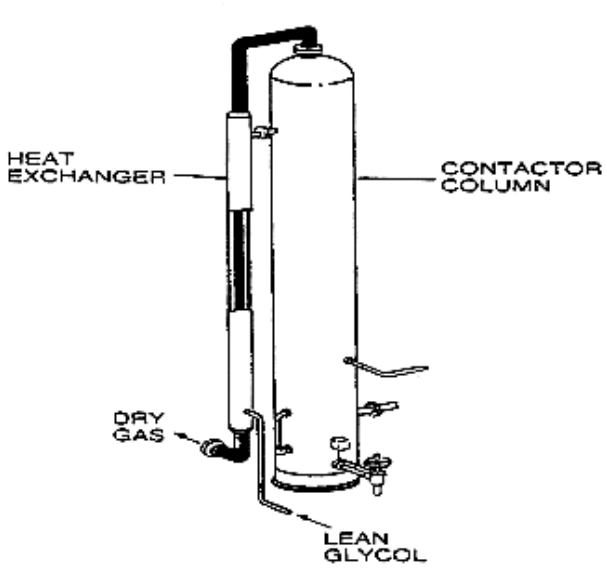


FIGURE 18 GLYCOL/ GAS HEAT EXCHAN JI R

On small dehydration units, this exchanger is often a double pipe heat exchanger, which is literally a pipe within a pipe. Lean glycol in the outer pipe flows around the outlet gas in the inner pipe. Larger units will utilise a shell and tube type of heat exchanger.

Heat exchangers are subject to fouling by salt, coke, or gum deposits which can reduce the heat transfer rate and increase the lean glycol temperature.

Glycol/Glycol Heat Exchanger(s): Hot, lean glycol leaves the surge tank and passes through a glycol/glycol heat exchanger (Figure 19), where the lean glycol is cooled by the rich glycol stream leaving the filters downstream of the G-C-G separator. A second glycol/glycol heat exchanger heats the rich glycol which leaves the reflux coil by the lean glycol stream leaving the first glycol/glycol heat exchanger.

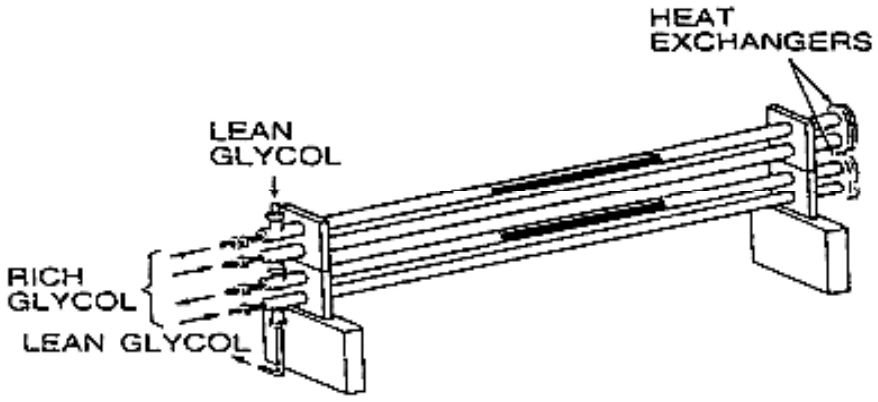


FIGURE 19 GLYCOL/GLYCOL HEAT EXCHANGERS

Lean glycol, which leaves the surge tank at approximately 400°F (204°C), must be cooled prior to entering the contactor in order for it to absorb the maximum quantity of water from the gas. These heat exchangers cool the lean glycol to about 212°F (100°C). They are very important in the overall operating efficiency of the dehydration unit. These heat exchangers substantially reduce the amount of heat required by the reboiler.

Process Description

As shown in Figure 20 the wet gas enters the contactor through an inlet near the bottom. The gas, travelling upward in the contactor column, is forced through the openings below the caps and bubbles through the glycol.

During the bubbling process, the gas gives up water vapor to the glycol. As gas passes upward through each succeeding tray it becomes drier.

Before leaving the contactor, the dry gas passes through a mist extractor to remove any glycol that may be in vapor form. As the glycol particles collect and become heavier in the mist extractor, they drop back into the top tray and rejoin the glycol stream.

The dry gas then leaves the contactor and passes through a heat exchanger where it cools the dry glycol entering the top of the contactor column. The dry gas is then ready for transmission.

Dry glycol enters the contactor tower at an inlet near the top and flows across the top tray, then downward and across other trays.

A level of glycol is maintained on a tray by means of a dam known as a weir. This level is above the slots in the bubble caps so the gas is forced to bubble through the glycol.

The glycol flows over the weir through an opening known as a down comer and into the tray below. Maintaining the level of glycol on the next tray above the bottom of the downcomer prevents gas from bypassing the bubble caps.

As the glycol spills downward through each succeeding tray, it becomes saturated with the water it has absorbed from the gas and collects in the bottom of the contactor.

The wet glycol from the contactor then passes through a filter where any abrasive particles and tarry hydrocarbons are removed before entering the pump.

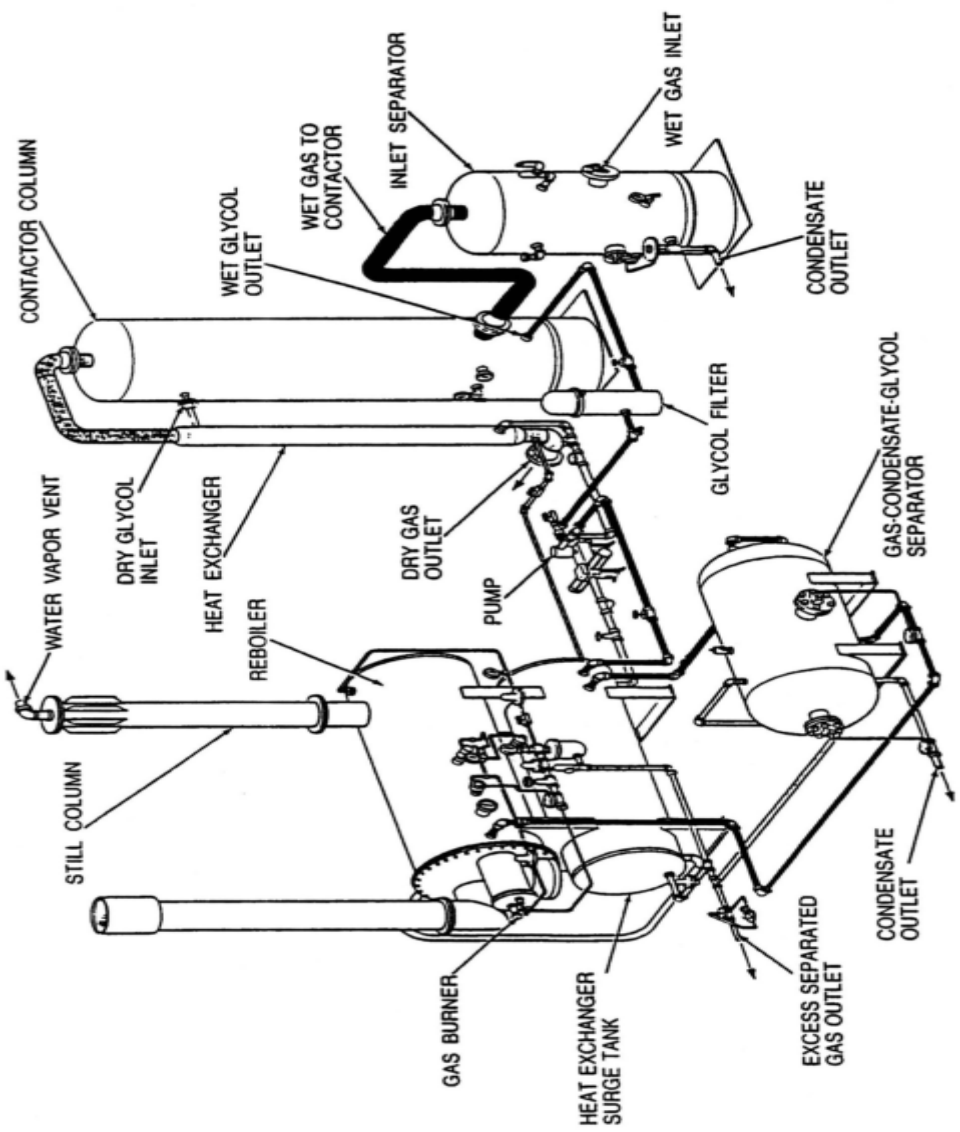


Figure 20 Glycol Dehydration Unit

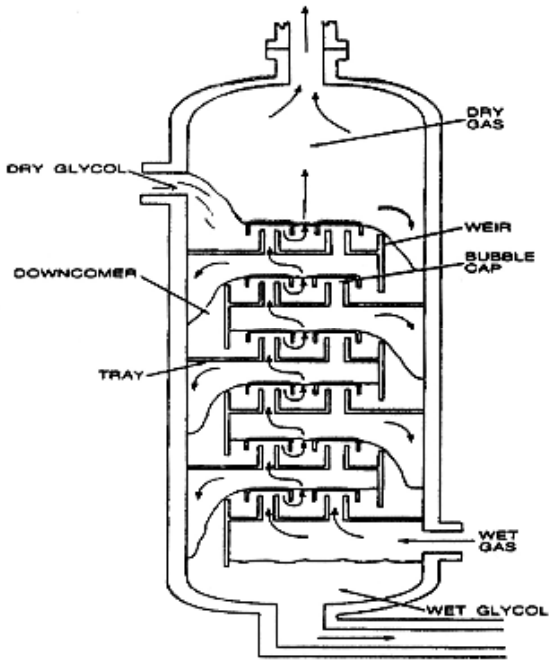


FIGURE 21 CONTACTOR COLUMN

From the pump, the cool wet glycol flows through coils in the surge tank.
This allows for a heat exchange process to occur. The cool wet glycol is warmed by the hot dry glycol before it enters the gas-condensate-glycol separator.

At the same time, the hot dry glycol is cooled before it enters the contactor column. From the coils in the surge tank, the wet glycol enters the gas-condensate-glycol separator.

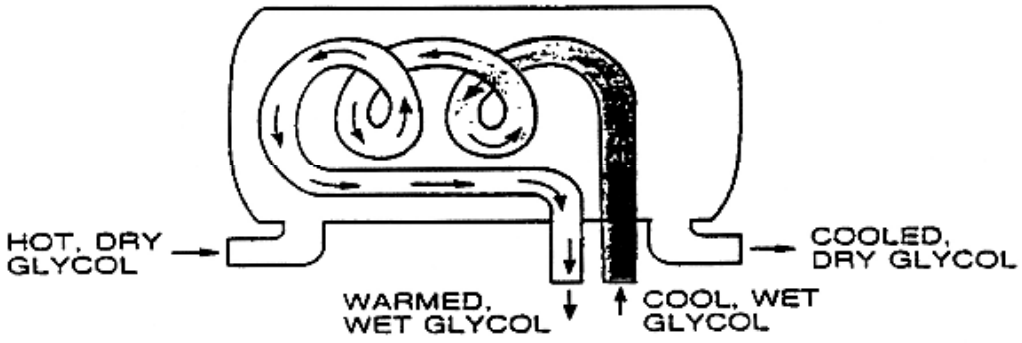


Figure 22 – Surge Tank Coils.

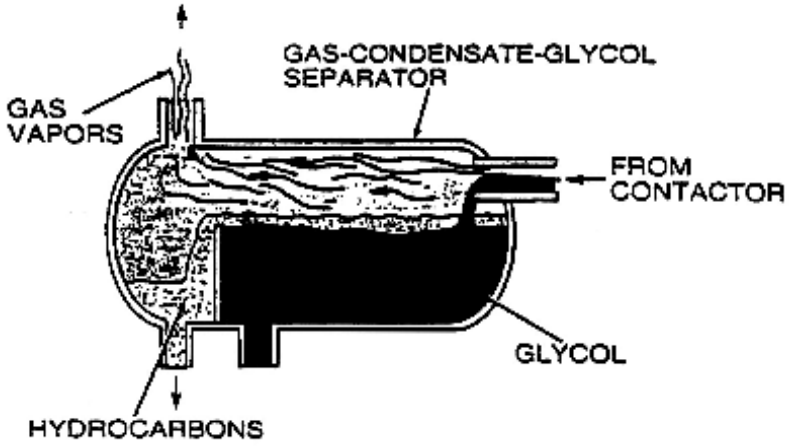


FIGURE 23 G-C-G SEPARATOR

Heat from the surge tank helps separate hydrocarbons from the wet glycol. The hydrocarbon condensate is skimmed off the glycol and any remaining gas vapors leave from the top.

From the gas-condensate-glycol separator, the wet glycol flows through a tube in the reboiler.

Here the glycol is heated to vaporise the water before entering the still column ("stripper") on the reboiler. The still column removes the water vapor from the glycol.

Inside the still column is a section, known as a Packed column, filled with ceramic, stainless steel or carbon packing known as saddles or rasching rings. The glycol spreads out uniformly over the packing and drips down through the lower portion of the packed column. The water vapor rises to the top of the column.

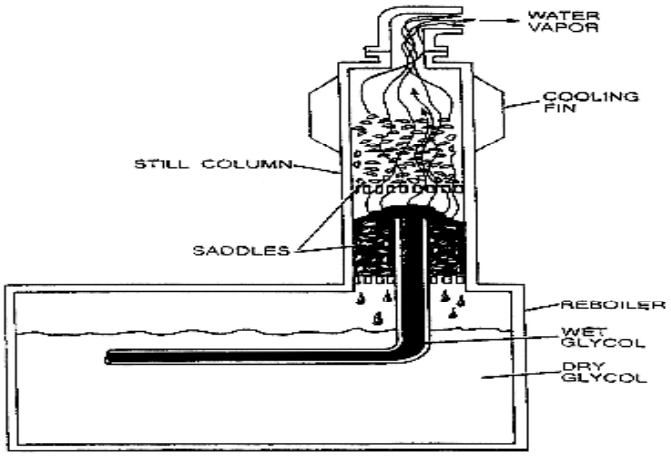


FIGURE 24 STILL COLUMN

From the packed column , the wet glycol drops downward into the bottom of the reboiler. A source of heat circulated through a tube in the lower section of the reboiler maintains the temperature of the glycol solution at approximately 370^0 to 400^0 F (188 to 204° C) which is just below the boiling and decomposition point of Triethylene Glycol. Waste heat from compressor or generator exhaust gases can be used as a heat source, but many installations also use a gas fired heater. The temperature of glycol in the reboiler is critical and must be controlled within this range.

The remaining water boils out of the glycol solution and moves upward through the still column as vapor. Some hot glycol vapors also are mixed with the water vapor. As this mixture passes upward through the still column, it comes in contact with a cooler part of the column and the glycol vapors are condensed and dropped back down into the reboiler. The water leaves the top of the still column as vapor.

The glycol level in the reboiler is maintained above the heating tube by the location of the overflow tube. The dried, purified glycol spills into the overflow tube and flows into the surge tank.

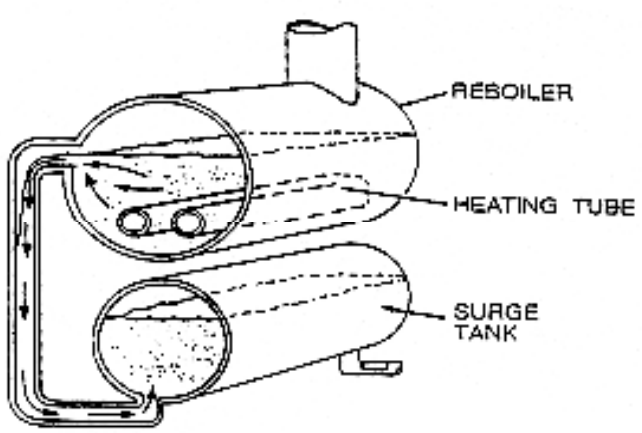


FIGURE 25 OVERFLOW TUBE

From the surge tank the dry glycol flows to a pump. The pump raises the pressure of the dry glycol slightly above that of the contactor column.

This dry glycol then passes through a heat exchanger which cools the glycol to near the temperature of the natural gas in the contactor. Proper temperature and pressure must be maintained in this system to prevent foaming.

With the return of the glycol to the contactor column, the dehydration cycle is completed and another cycle begins.

Unit Operation

Start up

Prior to the initial start up of a new plant, the vessels and lines should be thoroughly washed out with water to remove debris and corrosion products that accumulated during construction. After the system has been cleaned, start up is accomplished in three phases:

1. Establish glycol circulation throughout the plant.
2. Apply heat to the reboiler and bring it up to operating temperature.
3. Open the wet gas stream to the contactor and begin dehydrating the gas.

In order to circulate glycol throughout the system, it will be necessary to pressurize the vessels in the system. Pressuring can be done with wet gas or dry gas. The contactor pressure should be raised to at least 10 bars (150 psi) and the flash tank pressure should be raised to at least 3 bars (45 psi). When the vessels have been pressured, start up procedure is:

1. Fill the reboiler and surge tank with fresh glycol solution Also add to the flash tank.
2. Pressure up the contactor column by very slowly opening the gas inlet valve.
3. Prime and start glycol pump.
4. When liquid appears at the bottom of the contactor put the bottom level controller in service so the glycol will flow to the flash tank.
5. Put the flash tank level controller in service when liquid appears in the bottom, so that liquid will flow to the stripper.
6. Keep surge tank level half full by adding glycol when needed.
7. When desired circulation rate is established, light the reboiler or put the heat source in service and slowly bring reboiler temperature up to 250°F (121°C). Leave temperature at 250°F (121°C) until all water has been boiled out of glycol.

Recommended steps for safely lighting a gas-fired glycol dehydration unit:

a. Close pilot valve, gas burner valve, and main gas supply valve.
b. Be sure all gas is vented from firetube and flame arrestor is in place. Make sure gas is off five minutes prior to lighting.
c. Be sure firetube is covered with fluid.
d. Stand to one side while inserting burning torch. This protects the operator from ignition of residual vapors.
e. Adjust torch position to ignite pilot.
f. Open main Supply valve.
g. Slowly open Pilot valve.
h. Open burner valve after pilot is lighted and torch removed.
i. Adjust gas and air mixture to obtain proper flame.

j. Set fuel pressure as low as possible to maintain proper temperature.
8. Slowly switch gas stream through contactor column by opening the outlet line and slowly closing the by-pass line.
9. Slowly raise the reboiler temperature to around 380°F to 400°F (193°C to 204°C). Inspect unit for leaks and other abnormal operating conditions. Check gas dew point after the unit has been operating for about four hours.

Routing Operation

Routing operating checks include the following:
1. Check levels in each vessel and reset level controller as necessary.
2. Check the pressure drop across the filter and replace the elements as required.
3. Check the temperature of the lean glycol out of the glycol exchanger to see that the proper transfer rate is occurring in the exchanger.
4. Check the flow of glycol to the contactor and of stripping has to the reboiler.
5. Check the pressure of the flash tank to see if it is at proper level.
6. If water or air is used to cool the glycol prior to its entry into the contactor, check the glycol temperature in order to ensure that it is about 5°C to 7°C (10°F to 15°F) above the inlet gas temperature. Adjust the flow of air or water through the cooler as required.
7. Check the water content of the outlet gas to see that it is below the design limit.

Shut Down

This procedure is used to shut down a glycol plant:
1. Block gas to contactor column,
2. Shut off heat (leave pump running),
3. When unit cools to safe temperature, 150°F (66°C), shut off pump.
4. Drain glycol, if necessary.

Design and Operating considerations

The degree of water removal is determined by several process variables which include: inlet gas temperature, inlet gas pressure, gas flow rate, inlet glycol temperature, glycol concentration, and glycol circulation rate. All are major factors which affect the efficiency of a glycol dehydration system.

Inlet gas temperature

Incoming gas temperature has a significant effect on the water content of the gas entering the contactor. At constant pressure, the water content of inlet gas increases as the gas temperature is increased. If in contact with free water, gas will absorb additional water vapor as the temperature is increased. If the gas contains more water vapor, the unit is required to remove more water in order to achieve the desired water content of outlet gas. Example: At 1000 psig (6900 kPa) inlet pressure, changing the gas temperature from 100°F to 120°F (38°C to 49°C) will double the

amount of water vapor that will be carried by saturated gas. Inlet gas temperatures above 130°F (54°C) make normal glycol dehydration almost impossible.

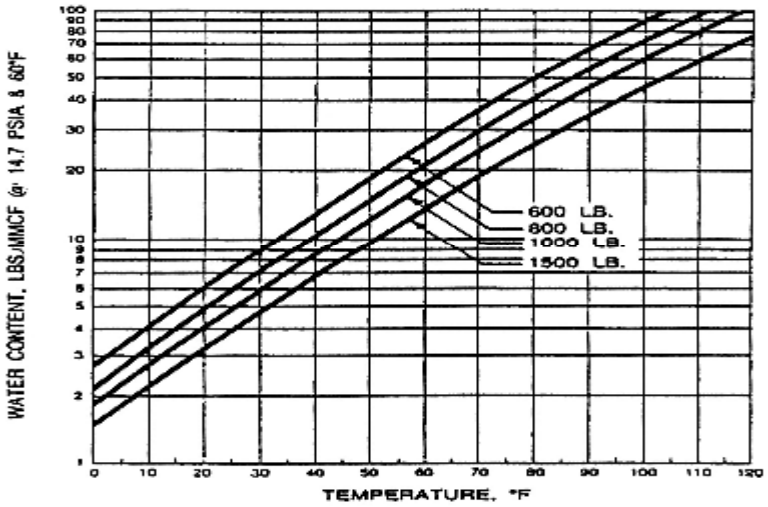


FIGURE 26 WATER VAPOR CONTENT OF NATURAL GAS AT SATURATION

In cold weather, line heaters are used to heat the gas stream ahead of the dehydration unit to maintain minimum inlet gas temperature, which is 50°F (10°C). Minimum inlet gas temperature is set by the viscosity of the glycol being used. Depending on the composition and pressure of the gas, hydrates may form at temperatures well above 50° F (10° C), thus requiring a higher inlet gas temperature to avoid plugging problems.

Low temperatures increase the tendency of glycol to foam which can create a significant operating problem.

Inlet Gas Pressure

In the normal operating range of a glycol dehydration unit, pressure is not a critical factor. However, it is important to note that at constant temperature, gas can hold more water as the pressure is reduced. Therefore, water content of inlet gas will be high if the gas pressure is low.

Contactor Pressure

Operating the contactor below design pressure of the system creates the following problems:

- At constant temperature, water content of inlet gas increases as pressure decreases, causing the unit to work harder to dry the gas
- At constant gas rates, the gas velocity through the contactor increases as the pressure is lowered, causing carryover problems

Gas Flow Rate

For a unit to operate efficiently, a specified range for gas flow must be maintained. If flow rate deviates outside these limits, problems can develop. Falling below the specified range will result in loss of efficiency and may result in increased dew point for outlet gas, depending on the design of the trays or packing in the contactor column.

If the flow rate rises above the specified range, reboiler overload occurs resulting in insufficient glycol regeneration and increased outlet gas dew point. The amount of glycol carried out with the dry gas will also increase. In fact, all the glycol can be carried out of the contactor column at one time if the gas flow rate is high enough.
A flow rate without rapid surges or changes is necessary to prevent a loss of seal in the contactor column downcomers. Seal loss allows gas to bypass the trays and results in high outlet gas dew points and high glycol loss with the outlet gas.

Inlet Glycol Temperature

Inlet glycol temperature of 10-15°F (5-7°C) above the inlet gas temperature is recuired to prevent hydrocarbon condensation in the glycol. Hydrocarbon condensation increases foaming, reduces carbon filter life, and increases carbon build up on the reboiler firetube.

Glycol Circulation Rate

If glycol circulation rate is increased without changing other variables, the outlet gas dew point can be decreased. However, it is desirable to maintain the lowest glycol circulation rate possible to produce the desired outlet gas dew point.

An excessive glycol circulation rate can result in the following problems:

- Increased fuel consumption
- Exceeded reboiler capacity
- Increased overhead glycol loss due to the higher temperature of lean glycol going to the contactor column
- Increased pump maintenance

Glycol Concentration

Changing the degree of glycol reconcentration (usually referred to as glycol purity) from the regenerator will produce the largest effect on dew point depression. Example: Assuming a contactor has six trays and a glycol circulation rate of three

gallons per pound (24 litters per kilogram) of water vapor in the inlet gas, the highest dew point depression obtainable with 98.5% pure triethylene glycol is 67°F (19°C). Changing the concentration to 99. 1 %, the dew point depression becomes 75°F (24°C). Higher dew point depressions can be obtained with higher glycol concentrations ; however, stripping gas must be used to achieve these levels of glycol purity.

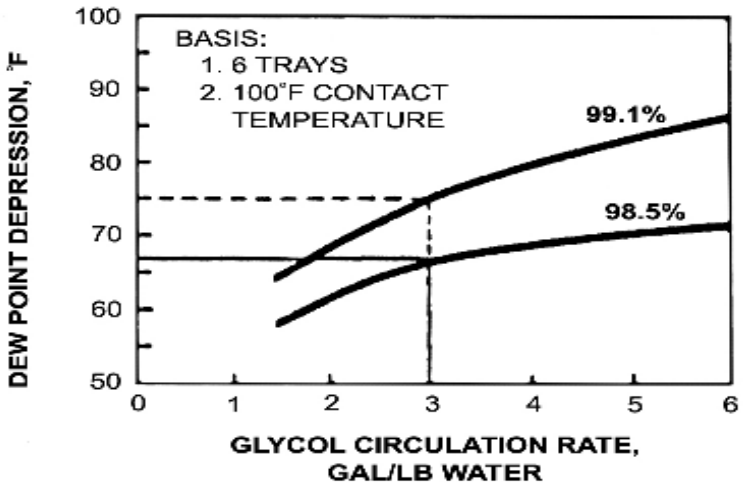


FIGURE 27 DEW POINT DEPRESSION VS. GLYCOL CIRCULATION RATE

Stripping gas is natural gas percolated through glycol being regenerated to help remove water that cannot be removed by the distillation process. In some systems, the gas is percolated through the glycol inside the reboiler. In other systems, there is a stripping column located between the reboiler and the surge tank.

Reboiler Temperature

This temperature controls the concentration of water in lean glycol; higher temperatures at constant pressure results in greater glycol purity. The range should be 380°F to 400°F (193°C to 204°C) for TEG. The temperature should NOT exceed 404°F (207°C) or glycol degradation will result.

Still Overhead Temperature

The still over-head temperature in the top of the still is very important. However, it is fairly easy to maintain an acceptable range for separation of water and glycol due to the wide difference in the boiling points of water, 212°F (100°C), and triethylene glycol, 546°F (286°C). The recommended range for the still overhead temperature is about 212°F to 215°F (100°C to 102°C) at sea level.

A temperature too low can cause reboiler over-load and excessive fuel consumption. A temperature too high causes glycol loss due to vaporisation.

Regenerator Pressure

A build-up in regenerator pressure, which is normally atmospheric, usually indicates a plugging or flooding problem, most commonly in the still packing. An increase in regenerator pressure also prevents some water from boiling out of the glycol, requiring higher temperatures in the reboiler, which wastes fuel gas and can lead to degradation of the glycol.

Filter Pressure

A pressure drop of 3-6 psig (21-41 kPa) is normal as glycol flows through the elements; however, the pressure differential increases as the elements become plugged. When this differential increases to 15-20 psig (104-138 kPa), the element should be replaced.

G-C-G Separator Pressure

Operating pressure in the G-C-G separator is 50-85 psig (345-587 kPa). Back pressure is maintained by a pressure controlled regulator.

Heat Exchanger Temperature

Glycol temperatures taken both before and after passage through a heat exchanger indicates the condition of the exchanger surfaces. As the exchanger surfaces become more severely fouled, the temperature change or differential across the heat exchanger decreases and energy consumption for glycol regeneration goes up due to increased firing of the reboiler.

Reducing Glycol Losses

The loss of glycol is a costly operating problem. Losses occur due to vaporization, entrainment, and mechanical leaks.
Typical locations for glycol losses are:

- Carryover with outlet gas leaving the contactor
- Vaporization with water vapor leaving the still column
- Leaks at the pump or pipe fittings
- Removal of liquid hydrocarbons and/or gas from the G-C-G separator

Conditions affecting glycol losses include:

- High still overhead temperature
- Contactor column operating at excessive gas rates
- Foaming
- Rapid changes in gas rate.

Note: Losses not exceeding one pound of glycol (or 0.1 U.S. gallon) per MMscf, of gas processed are generally acceptable.

Glycol Tests

In order to prevent costly operating problems such as high glycol losses, foaming, and corrosion, it is essential that meaningful analytical information be collected to pinpoint areas of inefficiency. Maximum drying efficiency is achieved when sufficient data from glycol test results is known so that proper evaluation of system performance and correct adjustments can be made.

Glycol Weight Percentage

This refers to the amount of glycol in the glycol solution. Lean glycol should contain about 98.5 to 99.9% a glycol. Rich glycol content varies from about 93-96% glycol.

Water Content

This refers to the amount of water in the glycol solution. Low water content in lean glycol is vital to good dehydration. The water content in lean glycol should not exceed 1.5 percent. A higher water content than 1.5 percent is usually an indication of a reboiler temperature that is too low and/or of a heater not operating properly. The water content in rich glycol should not be more than 5 or 6 percent. Water content spread refers to the difference in water content between lean and rich glycol. A spread which is too narrow (0.5 to 2 percent) usually means the glycol circulation rate is too high. The rate should be lowered to save energy, minimise glycol losses and improve dehydration. A spread over 4 to 6 percent is too wide and usually means the glycol circulation rate is too low and should be increased. A guideline for circulation rate is three gallons (11 litters) of glycol for every pound (0.45 kilogram) of water to be removed.

Hydrocarbon Content

This refers to the amount of oil, paraffin, or condensate in the glycol. It should be kept below 0.1 percent, otherwise it will cause problems such as foaming, plugging, or fouling which result in glycol losses and poor dehydration of the gas. Hydrocarbons get into glycol in two ways;

- Condensation which is caused when glycol enters the contactor colder than the incoming gas. This problem can be eliminated by maintaining a temperature for the entering lean glycol of 10-15°F (5-7°C) warmer than the incoming gas.
- Carryover of hydrocarbon contaminants from the inlet separator or gas

Salt Content

This refers to the amount of salt or chloride in the glycol. It should be kept below 0.01 percent by weight or 100 ppm.

Salt has a solubility in glycols of about 650 to 700 ppm. A higher salt level usually results in an accumulation of salt on the fire tube in the reboiler which leads to a decrease in heat transfer efficiency. Thus, it becomes more difficult to remove water from the glycol and poor dehydration usually results.

Salt contamination is a result of improper gas scrubbing; however, since salt usually travels in the gas stream in a fine mist, not even an inlet separator can effectively remove all salt. A coalescing or filter type separator is usually needed to remove salt from the gas stream.
Problems which occur when the salt content reaches almost 0.1 percent by weight or 1,000 ppm include:

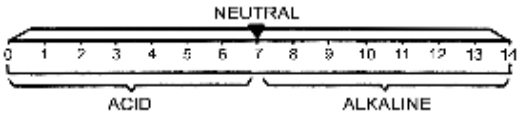
- Build-up of salt on the reboiler fire tube , resulting in decreased heat transfer efficiency
- Corrosion which will eventually decay the fire tube
- Hot spots on the fire tube , resulting in thermal decomposition of glycol and holes in the fire tube .

Solids Content
This refers to the amount of suspended solids in the glycol. Suspended solids should be kept below 0.01 percent by weight. This can be done by filtration.
Problems resulting from a high solids content include:

- Increased pump wear from abrasion
- Accelerated corrosion and erosion
- Increased fouling of fire tube .
- Increased glycol loss due to foaming
- Increased plugging problems F. pH

PH

This refers to the corrosiveness of the glycol, pH is a term used to express acidity or alkalinity. pH is based on a 0 to 14 scale.
Acidic = 0 to 6.9,
Neutral = 7.0
Alkaline = 7.1 to 14



pH SCALE

As glycol pH decreases, equipment corrosion rate increase rapidly and troublesome corrosive compounds result. Examples include:

- Organic acids from oxidation of glycol
- Thermal decomposition products
- Acid gases picked up from the gas stream, such as CO2 and H2S

Glycol pH should be checked periodically and kept on the alkaline side by neutralizing the acidic compounds with alkaline chemicals, such as monoethanolamine (MEA). A pH of about 7.3 is considered a safe level. Raising the pH above 8 to 8.5 is not desirable because of the tendency for an alkaline glycol solution to foam and emulsify more easily.

Iron content

This refers to the amount of corrosion products in the system, of which rust is the most common. Corrosion products either come in with the incoming gas or they are the result of a corroding glycol system. An iron content exceeding 10 to 15 ppm calls for corrosion prevention methods to be considered.

Operating Problems

The operating problems most frequently encountered in a glycol plant are:

- Foaming
- Degradation
- Corrosion
- Inadequate separation ahead of the absorber
- Water/ vapor in Gas
- Puking

All are inter-related to some degree. The usual symptoms of these problems are excess glycol make-up and / or lower normal dew point depressions.

Foaming

This is possibly the most common problem. Glycol tends to foam easily due to its viscosity. This tendency is promoted by liquid hydrocarbons (particularly aromatics), the presence of small solid particles, and contaminants. Poor tray design which uses too high vapor velocity through the bubble cap slots may be a contributing factor.

Drilling mud, corrosion by-products (iron sulfide) , glycol degradation products , and the like, promote formation of a stable foam.

The main problems associated with foaming are:

- Improper gas dehydration
- Excessive glycol losses when the glycol foams out of the contactor
- Additional equipment problems down- stream of the contractor from glycol carryover

To prevent glycol foaming, it is required:

- Use cartridge filter to keep glycol solution clean.
- An inlet gas separator to separate hydrocarbons entrainment.
- Use a foam inhibitor (trioctylphosphate is the most common foam inhibitor), too much foam inhibitor can promote foam.
- PH control are also very important.

Foaming losses will be minimized by proper mist eliminators, in most cases two in series at least 6-8 inches apart, are recommended . A third one can sometimes show advantage. The first unit is really a coalesce for the foam so that the subsequent unit (s) can separate properly

Degradation

Degradation is a natural occurrence and is accelerated in the presence of sulfur compounds. The answer is effective filtration. Degradation products contribute to foaming but they also are major sources of corrosion problems. The best filtration system uses a regular filter to remove the large "chunks" and then an activated carbon filter to remove hydrocarbons as well as fine contaminants that pass through the first filter.

Corrosion

Glycol will react with sulfur compounds to form undesirable degradation products "gunk" which is very corrosive and glycol decomposition products are primarily acidic. This condition can be controlled by good filtration and glycol reclaiming.

The PH of the glycol solution should be maintained from 6.5 to 8.5 by suitable materials, small amounts of base chemicals such as amines may be used.
Corrosion control chemicals used are:

a. Sodium mercaptobenzohiazole (sodium MPT) is usual used at 0.3 - 0.5 Wt.% concentration plus 1.5 Wt % dipotassium phosphate. The MPT concentration may be maintained by adding 1 quart of MPT to each 55 gallon drum of glycol.
b. Monoethanolamine (MEA) normally about 1 % in solution.
c. Coroval, usually used at about 40 PPm by weight. 4.4.4 Inadequate Separation

Salt is a continuing problem. Good separation ahead of the absorber is mandatory. Any salt arriving at the regenerator deposits either in the still column or in the reboiler. It is common for packed still columns to plug up to the point glycol is lost overhead. If this does not occur, salt can plug the reboiler and cause failure.

Water Vapor in Gas

The water vapor in gas is relatively fresh but is slightly saline. NaCl is soluble in TEG to some degree. At 50°C about 3.3 kg will dissolve in 100 kg of TEG. So, some salt is always present. The soluble salt hydrolyses to HCI and lowers the pH of the glycol.

Puking

It is almost always caused by a slug of liquid hydrocarbons entering the stripper. The hydrocarbons will flow down the stripper as a liquid, and when they reach the reboiler, they will vaporise almost instantaneously. The vapor flow up the stripper with sufficient velocity to carry out most of the liquid in the tower. Liquid hydrocarbons should be removed from the rich glycol in the flash tank, or a similar vessel.

General

- Good operating procedures require the blanketing of the system to prevent entry of oxygen into the system.
- Oxygen will cause the glycol to oxidise to acidic materials and enhance the corrosiveness of the system.
- Charcoal absorbers can be used to remove the heavier hydrocarbons which frequently accumulate in the system.

- Any neutralizer used to raise the PH will cause soluble iron in the glycol to precipitate and possibly plug the pumps and filters. The neutralizer should be added slowly and the filter system should be observed closely for the first few days of neutralizer use.
- Cocking of regenerator fire tubes caused by presence of solids or heavy hydrocarbons or by overheating can cause hot spots and tube failure.

Troubleshooting

PROBLEM	POSSIBLE CAUSE		CORRECTIVE ACTION
High Gas Dew Points	1. Insufficient Glycol Circulation	A.	Glycol powered pump. Close lean glycol discharge valve. Pump should stop. If pump continues to run, it needs repair. Repair pump.
•		B.	Electric or gas driven pump. Check circulation by closing rich glycol discharge valve from contactor. Time fill rate in contactor sight glass. Adjust circulation rate if needed.
		C.	Both types. Pump may be stroking but not pumping. Check valves for proper seating. Adjust or replace valves as indicated.
		D.	Both types. Check pump suction strainer for plugging. Clean strainer if needed.
		E.	Both types. Pump may be vapor locked. Open bleeder valve and bleed air.
		F.	Both types. Low level in surge tank. Add glycol to reboiler until proper level is restored in surge tank.
	2. Insufficient Glycol Regeneration	A.	Inadequate heat in the reboiler. Check temperature controller and fuel valve for proper operation. Verify reboiler temperature with test thermometer. Set at proper temperature if needed.
		B.	Check firetube for hot spots. Indication of coking or salt deposits on outside of tube. Pull firetube and clean if needed.
		C.	Check glycol/glycol heat exchanger(s) for leakage of rich gfycol into lean glycol stream. Repair or replace heat ex-changer(s) if needed.
		D.	Check reflux coil for leakage of rich glycol into the still column, if unit has a reflux coil. Pull coil and repair.
		E.	Still column plugged allowing rich glycol to channel through packing. Poor glycol/ vapor contact. Clean or replace packing if needed.

PROBLEM	POSSIBLE CAUSE		CORRECTIVE ACTION
	3, Operating Conditions O sfwvtThiafi Design	A,	Contactor pressure too low. Increase pressure to design pressure. May require installation of a back pressure valve.

		B.	Gas flow rate exceeding maximum system capacity. Compare actual flow rate to design flow rate. If actual is 10% more than design, system is probebly over loaded, Reduce gas flow rate back to design maximum.
		C.	inlet gas temperature above maximum design temperature. This allows gas to carry more water than system can re-move. Lower inlet gas temperature or flow rate.
		D.	Glycol circulation rate too low. Increase glycol circulation rate if possible,
High Glycol losses	1. Foaming	A.	Glycol tends to foam.when pH is above 8.5. Glycol becomes acidic when pH goes below 7.5. Check the pH and adjust to 7.0-7.5 with chemical neutralizer if needed.
		B.	Contamination of glycol with salt,liquid hydrocarbon, solid. or corrosion inhibitors lead to foaming. Remove source. Recharge system with fresh glycol if contamination too excessive
	2. Problems in Contactor C$tan	A.	Excessive gas flow rates creats turbulence which causes carryover of glycol with the gas stream. Check the gas flow rate and pressure in the contactor. Decrase flow and/or increase pressure on the contactor.
		B.	Plugged trays. When bubble caps become plugged, gas flow velocity increases throug remaining bubble caps rasulting in glycol carryover. Check trays and bubble caps. Clean if needed.
	3. Problems in Still Column	A.	Liquids entering system. Free water and liquid hydrocarbons may flood the still column resulting in poor glycol/vapor contact. Pressure surges may cause glycol carryover out the top with the vapor. Check inlet separator and/or filters. Perform appropriate maintenance.
		B.	Still packing plugged resulting in poor glycol/vapor contact. Overhead glycol carryover from excessive vapor flow. Clean

		or replace packing if needed.
	C.	High still overhead temperature results in loss of glycol vapor with the water vapor. Increase the flow of cool, rich glycol through the reflux coil. If still column has condenser fins, clean them to improve heat transfer.
4. Physical Leaks	A.	Entire system susceptible to leakage. Check all vessels, piping, fittings, pump packing, etc. Perform appropriate maintenance.

MODULE 4:

Overview of Sweetening & Sulfur Recovery

Overview of Sweetening and Sulfur Recovery

This is merely an overview of the subject of sweetening and sulfur recovery. A very complete summary of the subject is available in a companion book in the Campbell Petroleum Series, "Gas and Liquid Sweetening" by R. N. Maddox. The use of solid desiccants for absorption sweetening was reviewed briefly before.

The removal of sulfur compounds and carbon dioxide from natural gas continues to undergo change. The original purpose was to simply remove these "undesirable" components to prepare the gas for sale. The choice of process was made solely on the basis of initial and operating economics, since the cost of treating was a debit against gas sales revenue. Two factors have changed this situation--the higher discovery of H_2S and CO_2-bearing gases and an almost coincidental increase in the demand for sulfur worldwide. This changed the economic considerations. Now, the concern had to focus on the economics of gas sweetening plus sulfur production. The total package, rather than just sweetening alone, had to be considered. Old processes were reactivated, existing ones were modified, and new ones developed.

The engineer considering sweetening and/or sulfur recovery is faced with a wide choice of processes. Almost too wide the situation can easily be confusing. Each chemical vendor and engineering firm has one or more proprietary processes which they quite naturally tout. Furthermore, many of the processes are relatively unproven in practice by virtue of their newness. The operator is often faced with a decision between a tried-and-proven process like amine and a newer one which seems to offer advantage but is unproven. Is the potential economic advantage of the new process greater than the larger inherent risk involved?--a key question.

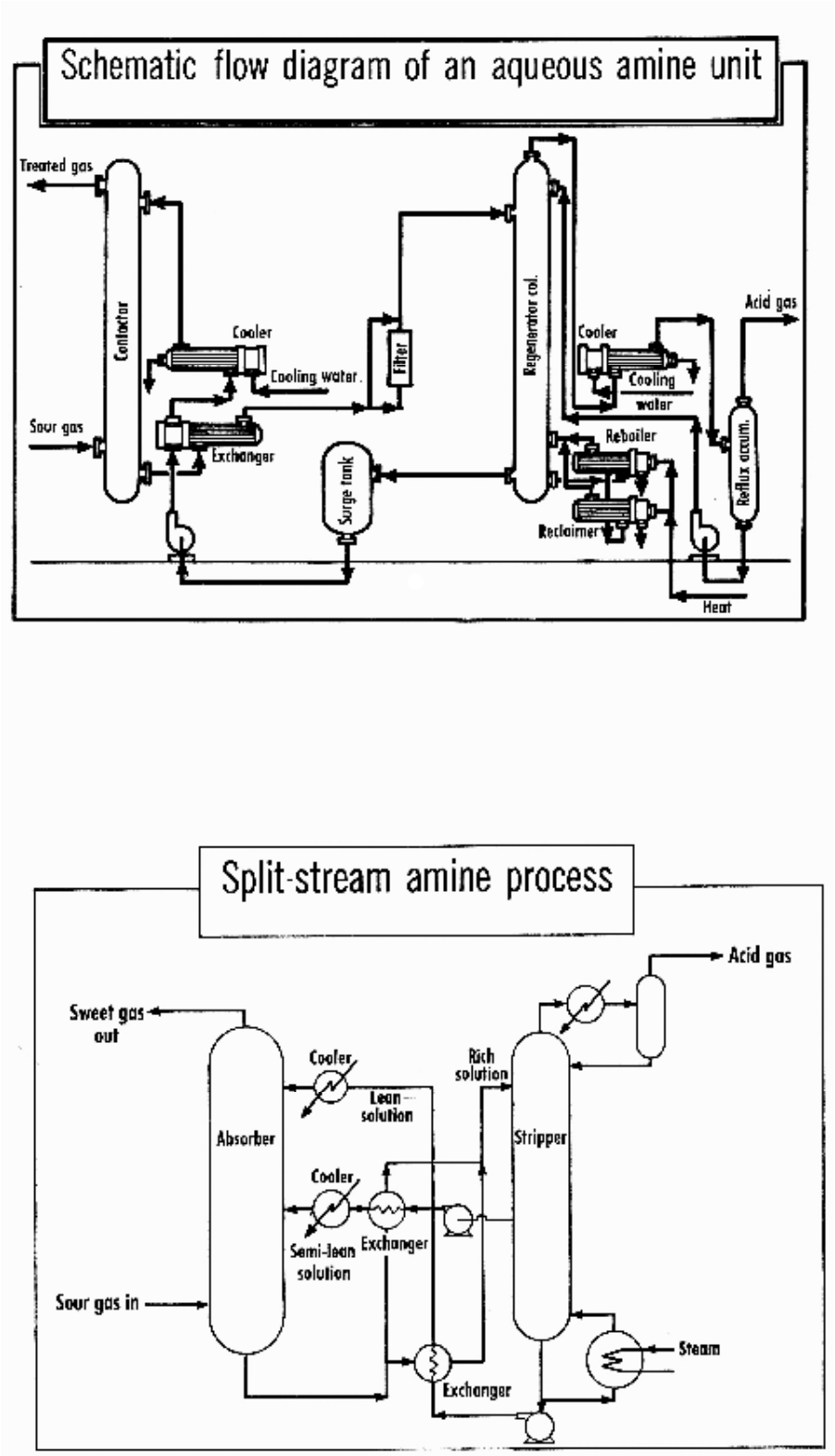
Hydrogen sulfide is the primary sulfur compound of concern (in terms of the quantity present), but carbonyl sulfide (COS), carbon disulfide (SC_2), and mercaptans are no less important. One basic design problem is obtaining a reliable analysis. Many of these sulfur compounds are present in trace quantities. Too often failure to analyze for them has led to the mistaken (and expensive) conclusion that they were absent. The story of those who fought sulfur compound problems with supposedly sweet gases is a long, sad one.

Carbon dioxide is not a particular problem in gas except for its reduction in the heating value. It is often removed for commercial use or for heating value reasons. For this reason, a process selective for H_2S (or CO_2) can sometimes offer advantage. Most processes offer some degree of selectivity by control of process variables but still remove both.

ETHANOLAMINE ABSORPTION

Figure 10.1 shows the basic flow scheme for those plants using one of the amines for absorption recovery of H_2S and CO_2- H2S is removed by a chemical reaction which is reversible by application of heat. C02 is physically absorbed. The flow sheet is almost identical with the glycol absorption plant and many of the same problems are encountered (foaming, corrosion) unless proper precautions are taken. Mono-ethanolamine (MEA) and diethanolarnine (DEA) are the most widely used solvents, particularly the former.

Figure 10.1 shows the simple system where lean amine enters the top of the absorber only. A split-flow system is sometimes used where circulation rates are high. In this system a semi-lean amine is withdrawn in the middle of the regenerator and enters somewhere in the middle of the absorber, where it picks up a large amount of acid gas on the bottom trays. The lean amine enters the top and makes the final separation. The two amine streams join in the bottom of the absorber and flow to the regenerator. This system is more expensive initially but has a lower steam cost.



Typical Absorption Sweetening Flow Sheets

The gas to be purified flows upward through a bubble tower contactor. An aqueous amine solution enters the top of the tower and flows downward from tray to tray, extracting the acid gas. Purified gas leaves the tower at its top. Foul amine, saturated with acid gas, leaves the bottom of the tower and passes through heat exchangers, in which the solution temperature is raised to approximately 82-93°C, and into the top tray of the stripper or reactifying still, which is also a bubble tower. This stripper contains in its base, or in an external kettle, a tubular heating element or reboiler. At the top of the stripper are condensers and a water separator.

The amine solution flowing downward through the stripper to the kettle is heated to approximately 110-116°C by hot vapors rising from the boiling solution in the kettle. Acid gas liberated from the amine solution and a portion of the internally generated stripping steam are carried overhead from the top of the tower through condensers where the steam is condensed and cooled. Condensed steam and cold acid gases are separated in the water separator, the condensate being returned to the top of the stripper as cold reflux, and the acid gases are burned in a flare or sent to a processing unit.

Hot lean solution stripped of its acid gas content flows from the kettle to exchangers where heat is exchanged with the cold foul solution. The stream then passes through water coolers where it is cooled to approximately atmospheric temperature. From these coolers the cold lean solution goes to a surge tank from which it is pumped to the contactors, thereby completing the cycle.

Because of the fairly rapid rate of reaction in the stripper, virtually complete stripping of hydrogen sulfide from the solution can be accomplished in a single tower.

Since aqueous amine solutions have very little affinity for hydrocarbon constituents of the gas to be purified, the process is applicable to the desulfurization of field gases prior to gasoline extraction, as well as to residue gas.
The reactions involved are very complex, but the following summarize the basic processes involved:

$$2\,RNH_2 + H_2S \underset{149°C}{\overset{38°C}{\rightleftharpoons}} (RNH_2)_2 \cdot H_2S$$

$$2\,RNH_2 + CO_2 + H_2O \underset{149°C}{\overset{38°C}{\rightleftharpoons}} (RNH_2)_2 \cdot H_2CO_3$$

149°C or

$$2\,RNH_2 + H_2S \underset{149°C}{\overset{38°C}{\rightleftharpoons}} (RNH_3)_2\,S$$

$$2\,RNH_2 + CO_2 + H_2O \underset{149°C}{\overset{38°C}{\rightleftharpoons}} (RNH_3)_2\,CO_3$$

Design Considerations

Inasmuch as chemical reaction is involved, the concentration of amine in the amine-water solution is one parameter. Also to be considered are solution circulation rate, amount and temperature of contact in the absorber, regeneration, proper heat exchange, and those factors necessary to minimize corrosion, solution losses, and utility costs.

In many cases the gas contract calls for reducing the H2S concentration in the exit gas to 0.6g/100 std m^3 [0.25 grains H_2S per 100 scf] of gas. A properly designed amine unit may easily meet this specification.

Corrosion, loss of amine, and operating upsets which prevent the delivery of specification gas are the major factors in amine plant operation. Consequently, minimizing these is the major factor governing process design. The following general "rules" (for specifications) will increase first cost but lead to optimum economics and improved operation:

1. The circulation rate of amine solution should be such that it will satisfy all three of the following criteria:
   a. The desired treated gas specifications can be attained using 4-5 theoretical trays in the absorber.
   b. The concentration of acid gas in the rich amine leaving the absorber will not exceed 0.3 mole acid gas per mole of amine.
   c. The temperature of the rich amine leaving the absorber should not exceed 49 C.
2. The lean amine entering the top of the absorber should be from 1-5°C warmer than the entering gas temperature in order to prevent hydrocarbon condensation.
3. The acqueous lean amine solution should not contain over 20% (by weight) MEA or 30% (by weight) DEA, whichever is used. 15% and 25% respectively are preferable if initial cost considerations will permit.
4. Avoid use of dissimilar metals to minimize bimetallic corrosion.
5. Provide an effective full-flow filter that is capable of removing finely divided iron sul-fide particles and "sludge" capable of removing all particles 5 microns and larger.
6. Regenerate the amine (operate the still) in accordance with the following:
   Max. pressure - 69 kPa (gauge)
   Max. reboiler temperature - 120°C
   Max. temperature of reboiler steam - 150 C
   Generate about 0.12 kg of steam per liter of amine solution circulated, in addition to that needed for reboiler heat. Lined packed towers, using porcelain packing, are preferred if size permits. If bubble trays are used, the slot velocity should be in the range of 3-5 m/s.

7. Amine solution velocities in the piping and heat exchangers should not exceed 0.6-1.0 m/s.
8. In the reboilers, tube spacing should be such that there is adequate vapor relieving capacity to minimize localized overheating and vapor binding.
9. If the above specifications are adhered to, carbon steel vessels and components are satisfactory, but stress relieving is necessary to minimize stress corrosion.
10. An amine reclaimer should be provided capable of processing one-half to one percent of the design circulation rate.
11. Provide an efficient inlet scrubber for liquid hydrocarbon.
12. Provide efficient mist extraction for the treated gas leaving the absorber.
13. The possibility of amine oxidation should be minimized by use of inert or sweet gas "blankets" wherever needed.
14. All control valves should have "soft seats." The valves controlled by the level control in the bottom of the stripper or reboiler should be downstream from the amine cooler. This prevents acid gases formed on expansion from causing corrosion in the amine-amine heat exchanger and lean amine cooler. The equivalent valve operated by the level control on the absorber should be downstream from the amine-amine heat exchanger for the same reasons.

MEA is preferred over DEA where degradation is not a problem because it has a higher absorption capacity, is cheaper, and the H_2S content of the treated gas may be routinely maintained at 0.6 g/ 100 std m^3 or less. However, MEA reacts irreversibly with carbonyl sulfide, carbon disulfide, and mercaptans to form degradation products. This cost for amine replacement can be excessive. Where such degradation has been a problem, the use of DEA has been common because of the almost total absence of such degradation. Based on past experience, however, it may be difficult to treat gas to less than 0.5 grains per 100 scf using DEA.

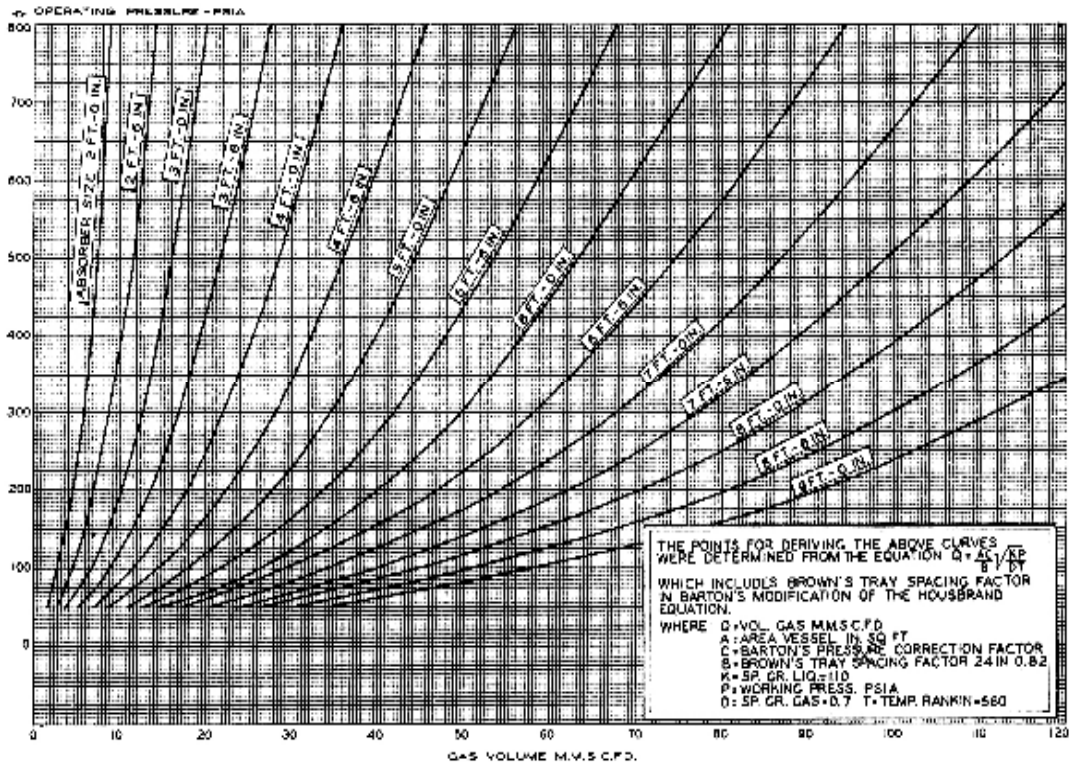
Absorber overall tray efficiency will be about 25-40%, so that 16-20 actual trays are usually used. With bubble cap trays a slot velocity of 3-5 m/s is a good maximum to minimize both foaming and erosion-corrosion.

Figure 10.2 may be used to estimate absorber size needed. Figure 10.3 is a comparable correlation for the regenerator (still) column. A two-foot tray spacing (0.6 m) is preferred because of potential foaming problems.

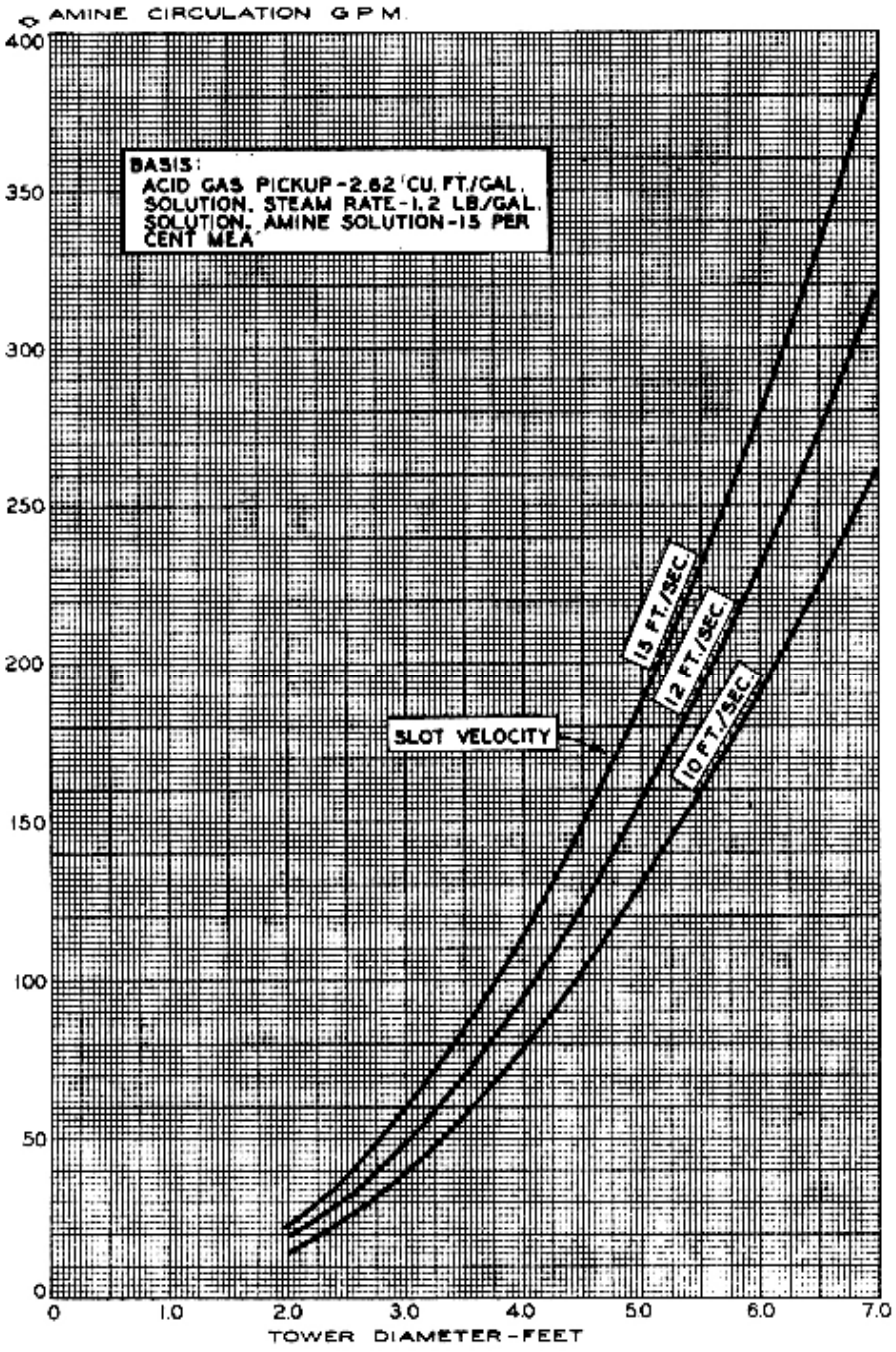
The inlet and outlet gas nozzles should be sized so that the gas velocity does not exceed the following:

Pressure	Gas Velocity
6.9 MPa [1000 psig]	1220 m/min [4000 ft per min]
3.5 MPa [500 psig]	2290 m/min [7500 ft per min]
0.3 MPa [40 psig]	3050 m/min [10 000 ft per min]

It is recommended that liquid velocity in the downcomers not exceed 12 cm/s [0.4 ft per sec]


OPERATING PRESSURE - PSIA
ABSORBER SIZE
2 FT.-0 IN.
2 FT.-6 IN.
3 FT.-0 IN.
3 FT.-6 IN.
4 FT.-0 IN.
4 FT.-6 IN.
5 FT.-0 IN.
5 FT.-6 IN.
6 FT.-0 IN.
6 FT.-6 IN.
7 FT.-0 IN.
8 FT.-0 IN.
9 FT.-0 IN.
10 FT.-0 IN.
THE POINTS FOR DERIVING THE ABOVE CURVES WERE DETERMINED FROM THE EQUATION Q = AC √(KP/(B(T-D)))
WHICH INCLUDES BROWN'S TRAY SPACING FACTOR IN BARTON'S MODIFICATION OF THE HOUSBRAND EQUATION.
WHERE Q = VOL. GAS M.M.S.C.F.D.
A = AREA VESSEL IN SQ FT
C = BARTON'S PRESSURE CORRECTION FACTOR
B = BROWN'S TRAY SPACING FACTOR 24 IN 0.82
K = SP. GR. LIQ. = 1.0
P = WORKING PRESS. PSIA
D = SP. GR. GAS = 0.7 T = TEMP. RANKIN = 560
GAS VOLUME M.M.S.C.F.D.

Variation of Absorption size with pressure and gas flow Rate



Sizing of Amine still columns

Amine Circulation Rate

Except for the absorber, all components may be related to the circulation rate of the amine solution. Therefore, establishment of this quantity is necessarily one of the calculations necessary.

The affinity of the amines for acid gases is a reversible chemical reaction. However, the chemical bonds are so weak that the general behavior pattern is similar to that of physical adsorption. The chemical equations generally used to characterize this reaction show that it requires two moles of amine to react with one mole of either CO_2 or H_2S. Conversely, the maximum capacity is 0.5 mole of acid gas per mole of amine. This represents the maximum equilibrium pickup or fixes the minimum circulation rate needed for completely treating a given quantity of acid gas. In a practical sense, this cannot be achieved because the absorbing solution is not completely stripped and absolute equilibrium cannot be achieved in an economic size of absorber. In essence, the design circulation rate is a compromise between this maximum and the economic limitations imposed by corrosion and economic problems.

Since the reaction governing H_2S removal tends to reverse at fairly low temperatures, the maximum absorber temperature should be not greater than 49°C (represented by the outlet temperature of the rich amine solution). In order to achieve this, the inlet sour gas should be cooled to at least 38°C before entering the absorber.

For the carbon dioxide reaction, contact temperatures may be as high as 66-71°C without seriously affecting efficiency. For this reason, when H_2S removal is involved, the circulation rate should be high enough to maintain the outlet amine temperature at 49°C or less.

Based on four theoretical trays, the usual design assumption, the circulation rate must be high enough to maintain the proper solution concentration in order that the desired mass transfer may be achieved. A series of absorption calculations has shown that this is not a factor when both the pressure and acid gas concentration are above the conditions which follow:

Abs. Pressure	Mol % Acid Gas
0.3 MPa [40 psig]	0.825
0.7 MPa [100 psig]	0.520
1.4 MPa [200 psig]	0.250
2.1 MPa [300 psig]	0.175

These values apply for absorption temperatures from 38-49°C. They show that gases do not have to contain very high percentages of H_2S and CO_2 to obtain effective mass transfer in this temperature range.

Extensive plant experience has shown that limiting the acid gas pick-up to 0.3 mol per mol MEA is a sound practice. This is far enough below the equilibrium value to be safe and furthermore minimizes corrosion and degradation losses. Loadings up to 0.4 mol acid gas per mole DEA have proven satisfactory. These limits enable effective regeneration and lead to reasonable amine consumption.

Plant experience has also shown that solutions containing 15-20% (by weight)MEA and 20-30% DEA represent a sound compromise between the cost of increased circulation rate and increased operating costs accompanying more concentrated solutions. Figures 10.4 and 10.5 provide an easy means of estimating circulation rate for these design criteria. The rate shown would normally be used for process design, providing the concentration and pressure were above the values shown above and outlet rich amine temperature specifications are met. The volumes of acid gas processed is simply the total volume of sour inlet gas multiplied by the mol percentage of H_2S and CO_2.

The following equation may also be used to estimate circulation rate:
m^3/min amine solution =

$$2.9\left[\frac{(10^6 \text{ std } m^3 \text{ inlet gas}) \times (\text{Mo1Fr acid gas})(\text{MW amine})}{(\text{mols acid gas/mo1amine})(\text{wt\% amine in soln.})}\right]$$

U.S. ga1/ min amine solution =

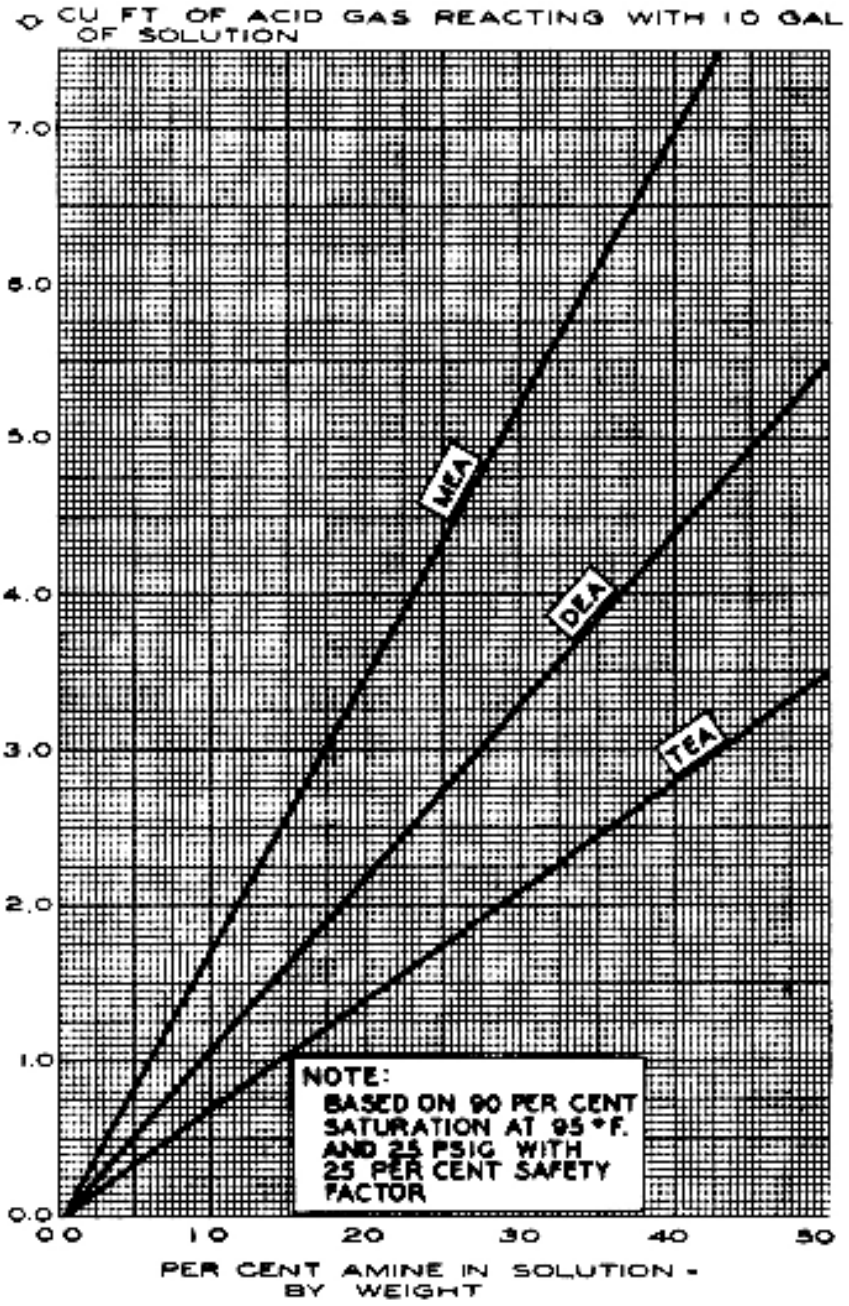
$$20.6\left[\frac{(MMscf/D \text{ inlet gas}) \times (\text{Mo1 Fr acid gas})(\text{MW amine})}{(\text{mols acid gas/mo1amine})\ (\text{wt\% amine in soln.})}\right]$$

MW MEA = 61.1 MW DEA = 105.1
The following conversion factors apply:

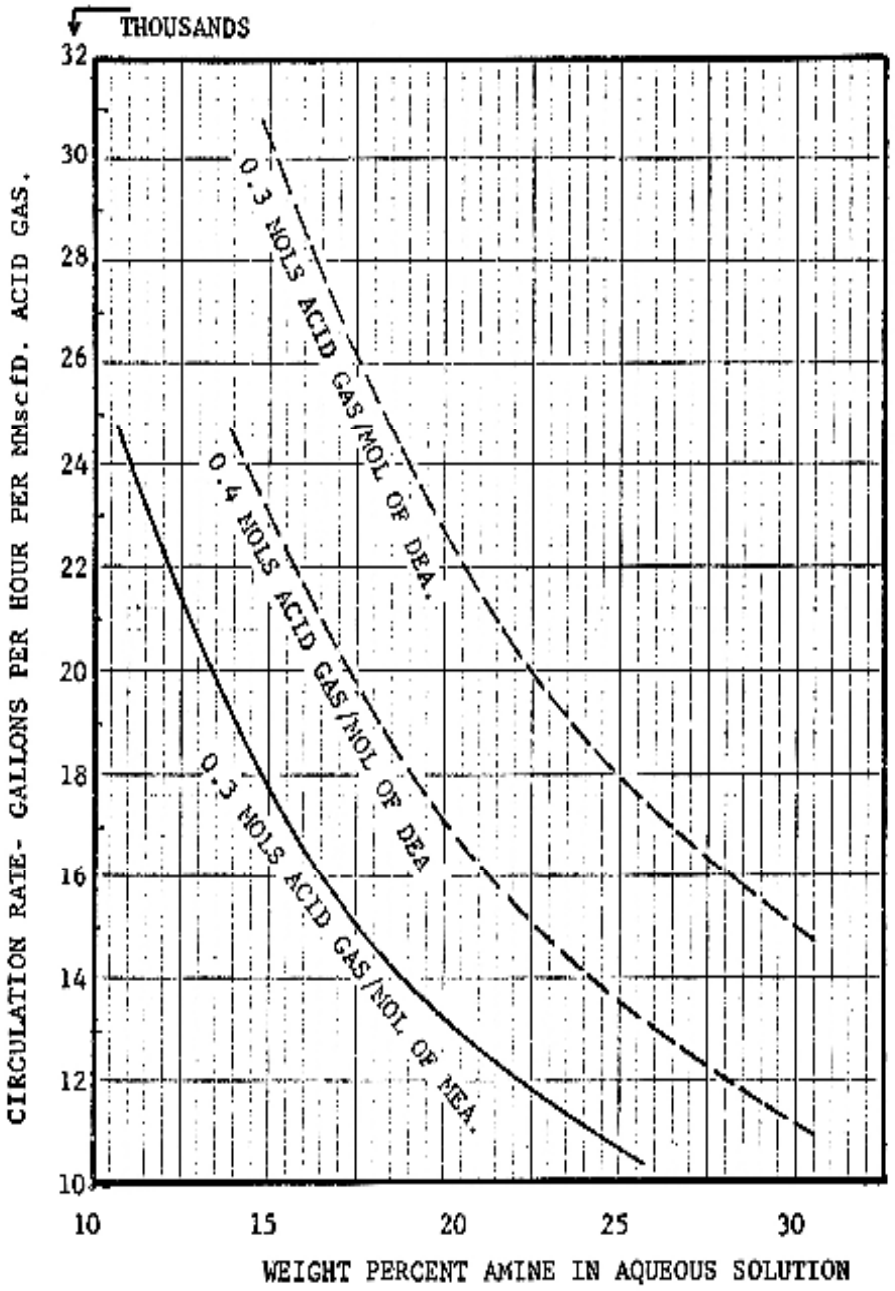
	Grains/ 100 scf	g/100 std m^3	Mo1 percent
H_2S	628	1500	1.0
CO_2	813	1943	1.0

Once a solution rate has been established, it is necessary to check whether the outlet amine temperature exceeds 49°C. For this purpose it is reasonable to assume that the treated exit gas will leave at the temperature of the entering lean amine, inasmuch as the majority of the absorption takes place in the lower section of the absorber. In may furthermore be assumed that the heat of reaction of H_2S and CO_2 with amine are as follows:

H_2S in MEA - 1907 kJ/kg
H_2S in DEA - 1189 kJ/kg



One Correlation for Acid Gas Loadings



Weight Percent in Aqueous solution

CO_2 in MEA – 1919 kJ/kg
CO_2 in DEA - 1519kJ/kg (To find values in Btu/1b divide KJ/Kg by 2.326)
A head balance around the absorber will yield the outlet temperature of the rich amine. In simplified form, the calculation may be carried out as follows:

Heat of Reaction

KJ/h= (10^6 std m^3/d) (Mo1 fr H_2S or CO_2) (C)
Where:

C = 1.19 (E+08) for H_2S in MEA	(If flow rate is in MMscf/d and heat of reaction is desired in Btu/hr, divide values of C show by 38.86 to make calculation.)
C = 7.40 (E+07) for H_2S in DEA	
C = 1.59 (E+08) for CO_2 in MEA	
C = 1.25(E+08) for CO_2 in DEA	

Heat Gain by Gas

KJ/h = 53 285 (10^6 std m^3/d) (Sp Gr gas) (Cp) (Δ tg)
Btu/hr = 3183 (MMscf/d) (Sp Gr gas) (Cp) (Δ tg)
Where:

	Metric	English
Cp = specific heat of gas		
Δ tg = Δ t between inlet gas and inlet amine	KJ/Kg. °C	Btu/1b. °F
	°C	°F

Temperature Rise of Amine

$$\Delta t_a(^\circ C) = \frac{\text{Heat of Reaction} - \text{Heat Gain of Gas}}{(4.2)\ (\text{liters/h of mine sol n. circulation})}$$

$$\Delta t_a(^\circ F) = \frac{\text{Heat of Reaction} - \text{Heat Gain of Gas}}{(8.4)\ (US\ \text{gal/hr of amine sol n. circulation})}$$

where: Δt_a = temp rise of amine solution in absorber
The last equation above may be solved for circulation rate to keep the outlet amine temperature below 49°C. For fairly sour gases where the restriction of 0.3 mol of acid gas per mo! of amine is stipulated, temperature rise is not normally a problem when the inlet gas is cooled to 38°C or less.

Regenerator (Still) Column

The size of this tower also may be related to circulation rate, even though vapor loading may be controlling. This is possible because the basic operation is substantially the same from plant to plant. Inasmuch as vapor loading normally controls, the exact size will depend on tray design. The maximum size of packed column used will depend on the vendor. A pressure of 50 kPa gauge was assumed in these calculations.

Between 75 cm and 90 cm diameter columns, vapor or liquid loading may govern depending on design. A downcomer velocity greater than 12 cm/s or a slot velocity greater than 5 m/s is not recommended.

Most columns have 15-20 actual plates or packing equivalent to 3-4 theoretical plates. As an approximation, it may be assumed that the reflux rate will be about 10-15 percent of the circulation rate. About a third of the trays should be above the feed tray to minimize amine loss.

Amine Solution Cooling

The total cooling load is obviously dependent on the circulation rate, reboiler temperature, and the inlet temperature of the amine solution to the tower. For a reboiler temperature of 116PC and an inlet temperature 1°C above the inlet gas temperature, Table 10.1 shows the total cooling load. The numbers shown are substantially independent of solution concentration and whether it contains MEA or DEA.

TABLE 20.1

Total Cooling Load for Amine Solution

Inlet	Gas Temp.	kJ/liter Amine Sol'n.	Btu/US gal Amine Sol'n.
27°C	[80°F]	371	1335
32°C	[90°F]	346	12A5
38°C	[100°F]	324	1165
43°C	[110°F]	301	1083

The total load shown above must be divided between the amine-amine heat exchanger and the solution cooler. In effect, the cooler inherits that portion of the cooling load not satisfied by the former.

The load in the amine-amine exchanger will obviously vary with many rather arbitrary design factors. Based on past plant performance, however, the values shown in Table 10.2 are suitable for cost estimation purposes. They are based on a 22°C approach in the exchanger and a 116°C reboiler temperature.

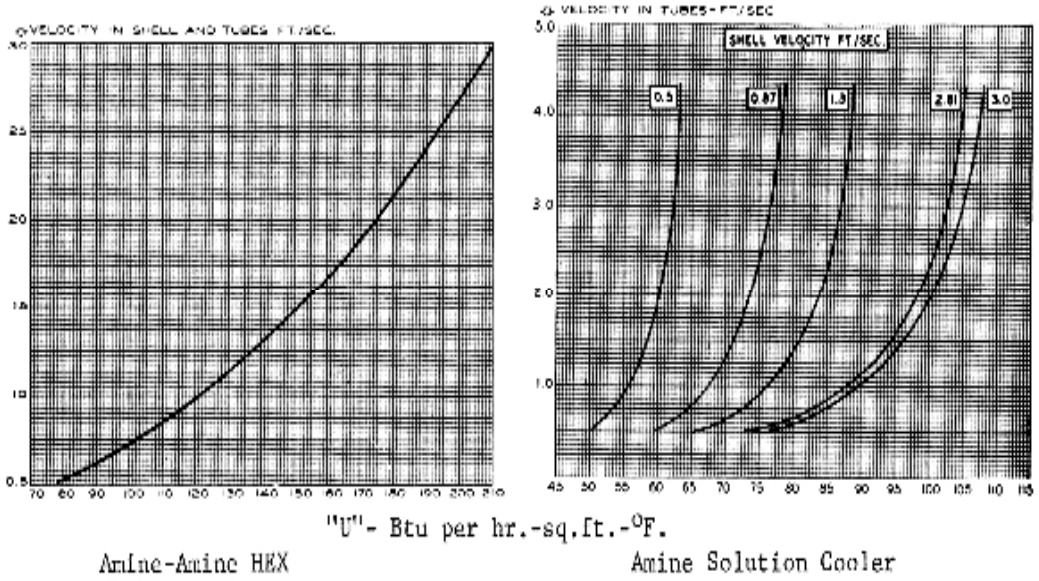
Table 10.2

Cooling (Heating) Load in Amine-Amine HEX

Temp, of Rich Amine from Absorber	kJ/liter Amine Sol'n.	Btu/US gal Amine Sol'n.
27°C [80°F]	278	1000
32°C [90°F]	254	915
38°C [100°F]	232	834
43°C [110°F]	209	750
49°C [120°F]	185	666

The resultant solution cooling load may be found from a combination of Tables 10.1 and 10.2.

The amount of surface needed to transfer the loads in question will depend on the cooling medium and the construction of the heat exchanger. For shell and tube exchangers, Figure 10.6 contains a useful correlation. For pipe-in-pipe units using finned inner tubes, an overall "U" of 674 kJ/h.m^2-°C [33 Btu/hr-ft2-°F], based on the finned (extended) surface, is suitable.



Overall Heat Transfer Coefficient for Amine Heat Exchange

If water is used to cool the amine, the circulation rate will depend on the temperature rise allowed. The specific heat of water is about equal to that of the amine solution, so as a general approximation the circulation rates can be assumed to be about equal. One can further assume that water makeup for this service will be about three percent of the circulation rate.

If air cooling is used, the amine cooler coils and the still column overhead condenser are normally incorporated into the same unit. Therefore, the total load must be found in order to estimate operating costs of this unit. There is no simple, reliable correlation for air cooling since one is balancing fan cost against surface cost for a given set of specifications.

Regeneration (Still) Column Heat Loads

The major factor governing cost is the amount of energy needed to power the reboiler. This depends on all of the design variables present. Steam requirements will usually vary from 0.11-0.18 kg of steam per liter [0.9-1.5 lb of steam per US gallon] of solution circulated.

The overhead condenser has to cool and condense the vapor leaving the top of the column to as low a temperature as economics permit. The vapor entering the condenser is largely composed of water vapor H_2S and CO_2. The quantity of hydrocarbon should be less than one percent (particularly for feed to a sulfur plant). A small amount of amine is desirable (less than 0.5%) in order to minimize corrosion. Since the load here is dependent on everything else that is specified, a complete heat balance would be needed. However, it has been found that the equation

Q(kJ/h) = (295)(liters/h of amine sol'n.)

Q(Btu/hr) = (1060) (US gal per hr of amine sol'n.)

yields reliable results for cost estimation purposes. This is based on the vapor entering at 96-107°C and leaving at 38-54°C. Most of the fluid condensed will be water. The condensation load accounts for about 75% of the total. As a general rule, the vapor from the still column should contain at least 3 pounds water per pound of acid gas, in high pressure natural gas service.

Physical Losses of Solution

Both amine and water are lost overhead from the absorber and still column. This loss will of course vary with design and operational practices. This loss is due both to the stripping action of the gas and the entrapment in the absorber. Said loss may be estimated as follows for cost estimation purposes:

Absorber at 38°C [100°F] or less

Amine Entrainment loss (kg/d) = (0.0049)(10^6 std m^3/d) [(lb/day) = (0.18)(MMscf/d gas)]

Water Entrainment Loss (kg/d) = (0.018) (10^6 std std m^3/d) [(lb/day) = (0.67)(MMscf/d gas)]

Amine Vaporization Loss (kg/d) = (0.027) (10^6 std m3/d) [1.0 Ib per MMscf/d gas]

The amine loss due to vaporization is calculated on the basis of available vapor pressure data. Entrainment losses are based on past experience where normal mist extraction devices were used.

The losses of amine and water from the still column may be estimated as follows:
Water (kg/d) = (68)(10^6 std m^3/d of acid gas) [(lb/day) = (2500)(MMscf/d acid gas)]
Amine (kg/d) = (0.07)(10^6 std m^3/d of acid gas) [(lb/day) = (2.5)(MMscf/d acid gas)]
In some plants the largest physical loss occurs by ".spillage." There is no way to estimate this accurately because of the human factor involved, but if said loss exceeds two percent of the total amine makeup required there is room for improvement.

Degradation Losses

In addition to the degradation due to temperature, both MEA and DEA react irreversibly with CO_2. The reaction is fairly slow but still accounts for a substantial loss of amine. The reaction is very complex, with side reactions, which makes it difficult to always accurately predict degradation losses. Not only do these products reduce the amine effectiveness for sweetening, but they also promote the corrosiveness of the fluid. Their presence usually requires the use of a side stream reclaimer.

Degradation of MEA is caused by CO_2, carbonyl sulfide, carbon disulfide, and mercaptans. DEA is only affected to any measurable extent by CO_2 and hydrogen cyanide. Based on limited data it is suggested that the following degradation rates be used:

MEA

2.6 kg per 10^6 std m^3 of CO_2 treated [210 Ibs per MMscf of C02 treated]
0.08 kg per 100 std m^3 of COS or CS_2 present [6.4 Ibs per 100 scf of CO_2 or CS_2 present]

DEA

2.8 kg per 10^6 std m^3 of CO_2 treated [230 Ibs per MMscf of CO_2 treated]
It has been reported that the degradation by COS and CS_2 may be virtually eliminated by the addition of sodium or potassium hydroxide to the amine solution.

Amine Reclaimer

The means used to reclaim amine vary somewhat in detail, but all involve the use of distillation to differing degrees. For cost estimating purposes, the average reclaimer will require about 90-170 Btu/U.S. gal of amine solution circulated, depending on design.

Figure 10.7 shows a typical gas-fired reclaimer. Usually 0.5-3.0% of the circulated volume is reclaimed.

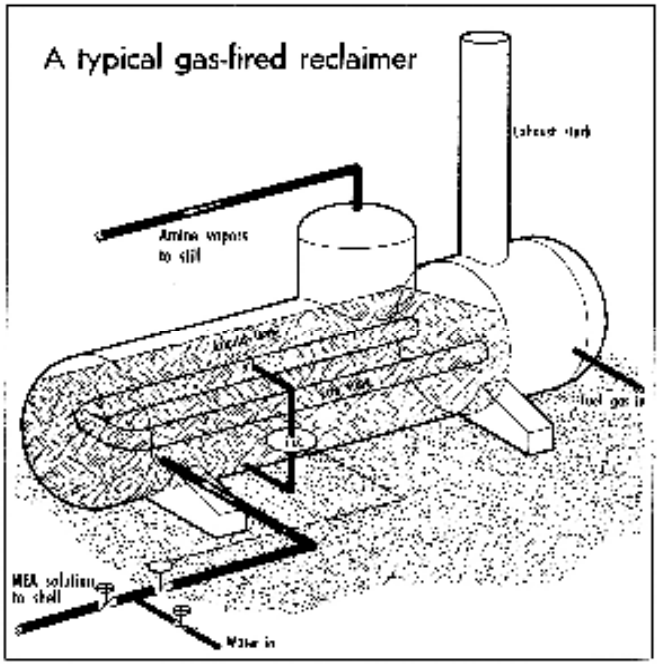
Filters

Any filter used should not pass a particle larger than 5 microns. Good filtration is one key to prevention of corrosion, for it minimizes particles which promote a stable foam and removes some of the degradation products. A full-flow filter should be used. Two are usually needed, for cleaning and replacing filter elements is routine. Some provision should be supplied to switch to another parallel element if pressure drop becomes too great. Paper and sock-type elements can fail and re-foul the system. If filters clog too often, this is a symptom of a problem somewhere in the unit. By-passing filters for relatively long periods of time to remove the nuisance of changing them is ill-advised.

Charcoal filtration on a cold lean amine stream is often successful, particularly when used downstream from a sock-type or pre-coat type filter. It adsorbs hydrocarbons and helps prevent foaming. In some instances, where desirable, charcoal regeneration is possible.

Operation of Amine Plants

The operation is often routine but requires close control on process variables. Good initial design subject to previous limitations is mandatory or continuous trouble is to be expected. Figure 10.8 shows the primary trouble spots.



Corrosion. Figure 10.8 also considers some aspects of corrosion based on the "NGPA Corrosion Detection Report." Shown are problems with sour gases in other than amine units. Corrosion results from sludge deposited on steel surfaces (by galvanic action), high velocity erosion-corrosion, and improper control of solution acidity. Stress corrosion is also a problem with improper stress relieving or poor choice of materials of construction.

Adding corrosion inhibitors is no solution to improper design, but it is often necessary. Carbon steel may be used throughout if proper control is provided. Types 304 and 316 stainless steel have been used at critical points. Type 7072 aluminum-clad bundles have been reportedly used successfully in heat exchanger service. Metal-clad and plastic-coated parts have also been tried. The success of the latter is controversial. Some report marked success and some do not. The key seems to be good metal-plastic bonding and prevention of "pin holes."

Addition of up to 7 grams per liter of sodium carbonate has sometimes minimized corrosion. Others have used hydrazine as an oxygen scavenger.

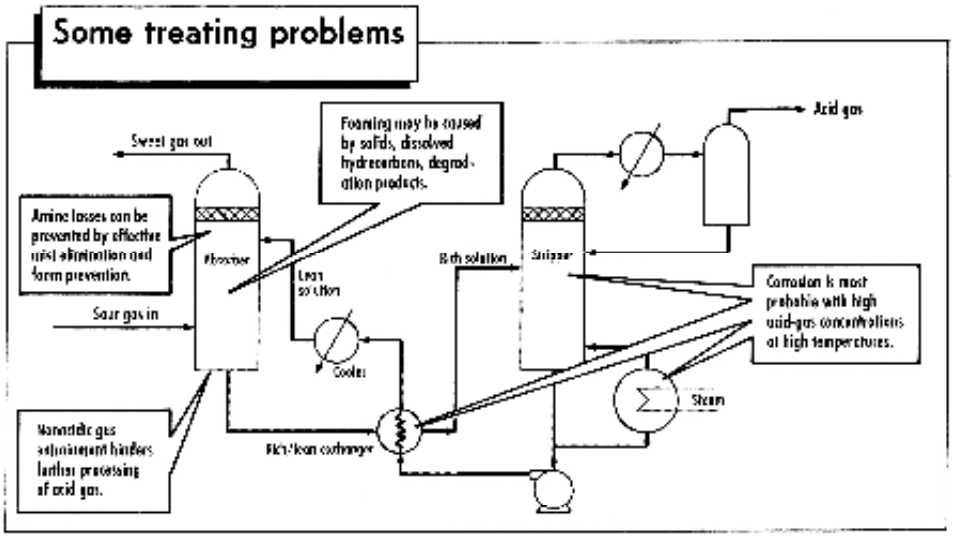
Because progressive embrittlement is a constant problem, some specify that control valve seats have a Rockwell hardness from 18-23, with 18-20 being preferred.

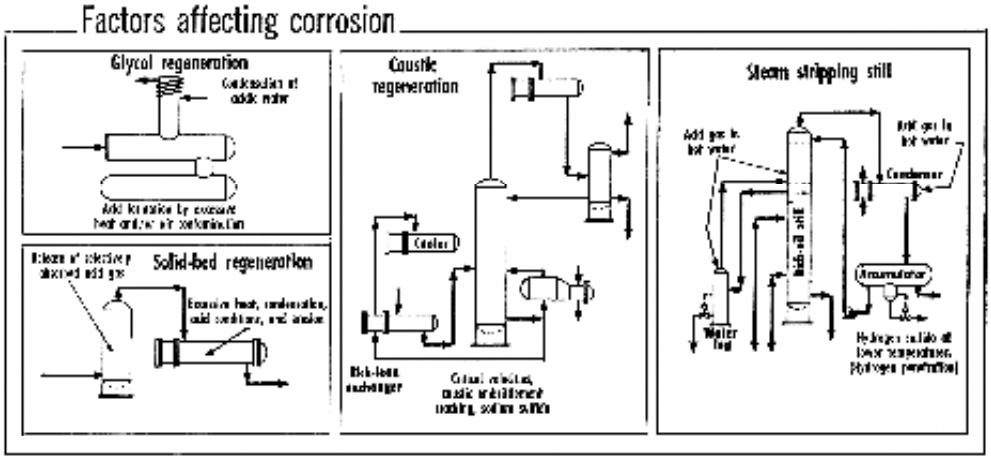
Foaming. Amine, like glycol, does not form a stable foam in a pure state. Comments made about foaming for glycols apply equally well to amines, including mist extractor design.

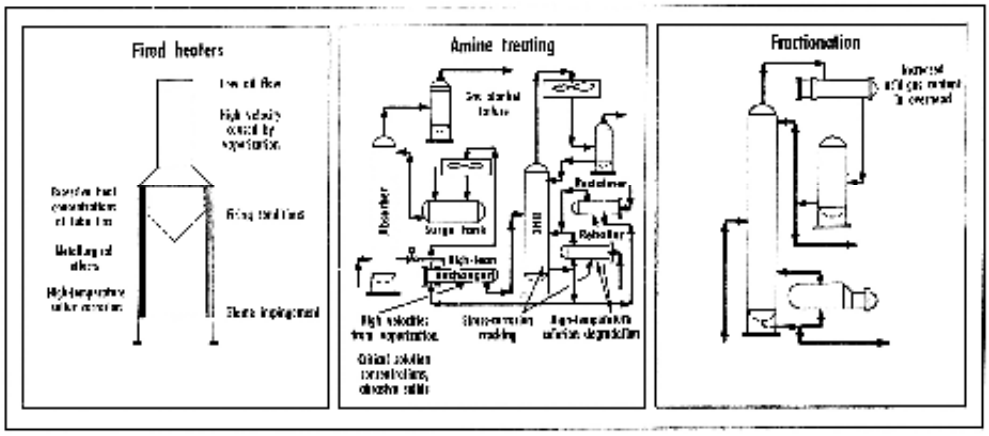
Caution must be used in choosing a foam inhibitor, which is (unfortunately) a trial-and-error process. Some materials which might help break an existing foam strangely enough can promote a stable foam if added before the fact. Some operators report success with materials like Oleyl alcohol, Pluronic L-61, Corexit 7669, and Ocenol. As noted previously, charcoal filtration often provides positive prevention.

SULFINOL

This plant has the same physical features as an amine plant. The only difference is the solvent. A Shall development, the Sulfinol solvent uses Sulfolane (tetrahydrothiophene dioxide) and an alkanolamine. One plant is reported to use a solution of 40% sulfolane, 15% water, and 45% diiso-propylamine. Its advantages would appear to be:


Some treating problems
Sweet gas out
Foaming may be caused by solids, dissolved hydrocarbons, degradation products.
Acid gas
Amine losses can be prevented by effective acid elimination and foam prevention.
Absorber
Lean solution
Rich solution
Stripper
Sour gas in
Cooler
Corrosion is most probable with high acid-gas concentrations at high temperatures.
Steam
Nonacidic gas entrainment hinders further processing of acid gas.
Rich/lean exchanger





Summary of Problems Points in Sour Gas Treating

1. A circulation rate about 50-75% of that used for MEA
2. Lower heat duty
3. About 1/8 the degradation rate of MEA
4. Reduced foaming problems
5. Efficient mercaptan removal
6. Steam consumption 'tends to be less.

In order to meet exit gas specifications, it would appear that the circulation rate should be such that the rich solution leaving the absorber contains no more than 0.04 std m^3 of acid gas per liter [6 scf of acid gas per U.S. gal] of solvent.
Carbonyl sulfide and mercaptans are removed by Sulfinol and there is no substantial degradation by COS. Typically, about 90% of the mercaptans will be removed when treating for H_2S and CO_2.

Sulfinol appears to offer economic advantage in at least some cases where the H_2S is larger than the CO_2 content of the gas.

DIGLYCOLAMINE (DGA)

This is a Fluor patented process marketed as the Fluor Econamine Process. It appears to offer promise when the stream contains a minimum of 2% acid gas. The primary advantage is a 15-20% lower capital cost and annual operating cost. The latter is a result of reduced steam or heat requirements. The process plant is essentially identical to one using amine or Sulfinol. DGA reacts with CS_2 and COS much like MEA. Allowable acid gas loading of the solution appears to be higher than for MEA. Since the allowable DGA concentration is higher, there is about twice as much acid gas removed per unit volume of solution. For a number of reasons, DGA is usually recommended only for gases containing more than 1% acid gas. DGA reacts with COS and CS_2 irreversibly. Reclaiming is thus necessary.

HOT POTASSIUM CARBONATE

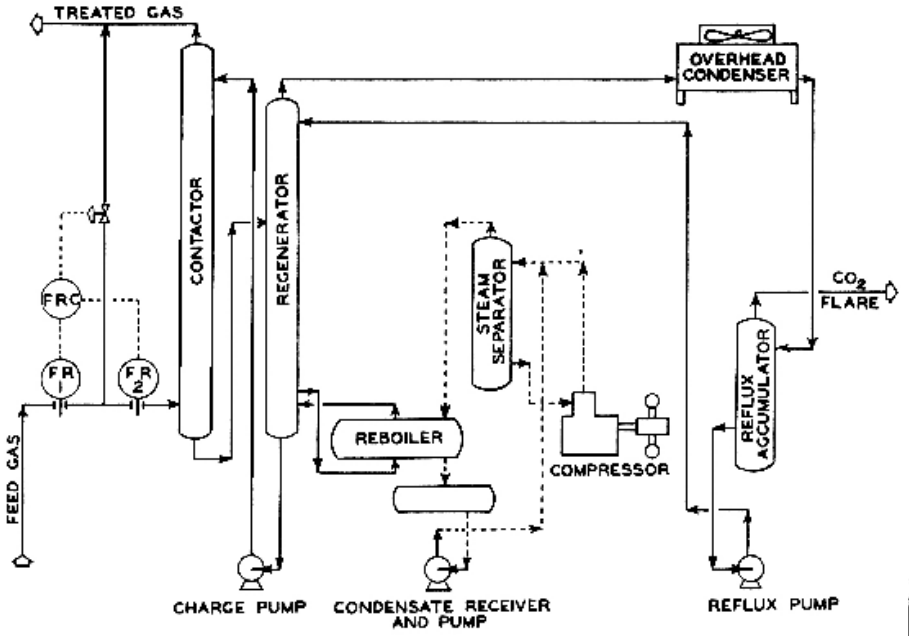


Figure shows the general flow sheet of the process. The general equations are:

$K_2CO_3 + CO_2 + H_2O \rightleftharpoons 2\,KHCO_3$

$K_2CO_3 + H_2S \rightleftharpoons KHCO_3 + KHS$

$CO_2 + H_2O \rightleftharpoons CO_2 + H_2S$

In the absence of CO_2, potassium bisulfide solutions are not readily regenerable so that this process is only good for H_2S when a reasonable quantity of CO_2 is present. As a rule of thumb, this process should be considered when the partial pressure of the acid gas is 1.5 bars or greater. It is not recommended for low pressure absorption or high pressure absorption of low concentration acid gas.

The process has the following advantages:

1. It is a continuous circulating system employing an inexpensive chemical.
2. It is an iso-therm system in that both absorption and desorption of the acid gas are conducted at as nearly a uniform high temperature as can be obtained, thus requiring no heat exchange equipment in the fluid circulating system.
3. The desorption of stripping is accomplished with a smaller steam rate than required for an amine plant.

The process is not without its disadvantages:

1. It will not economically reduce the hydrogen sulfide content of the gas stream to so-called pipe line specifications. To accomplish this quality, a conventional amine plant is used to "polish off" the residual left by the carbonate plant, after the gas stream has been cooled from the 93°C or hotter carbonate absorber before entering the amine contactor.
2. It is similar to other acid gas removal processes in that it is also prone to corrosion.
3. Like other liquid absorbents in sweetening plants, it is liable to the problems of suspended solids and foaming.

The process finds its economic niche where reasonably large quantities of excessively high concentrations of acid gas must be removed. Under an evaluated balance of work load between the carbonate and the necessary amine plants, the resulting combined plant investment and operating expense will be considerably below that of an amine plant by itself.

The circulation rate of the carbonate plant needs careful evaluation as in the amine plant. Basic equilibrium data indicate that as high as a 40% solution can be used, but these data do not consider the precipitation problem after possible cooling of the nearly saturated foul solution. A concentration of about 20% is adequate to pick up between 0.025-0.028 m^3 of acid gas per liter [3.5-4.0 ft^3 of acid gas per gallon] of solution without adverse results.

In stripping the foul solution, about 33-50% of the steam is required to strip an amine solution (containing less acid gas).

The carbonate solution must be protected from incoming liquid hydrocarbons, dust, and other suspensions that cause foaming. Suspended solids in the solution can be filtered out in charcoal filters or other high retention units. By keeping foam-forming additives out of the system, the carry-over from the contactor can be held within reasonable bounds.

By the precautionary use of stainless steel in a few spots, such as at the solution control valve out of the contactor, in the line downstream of that valve to combat two-phase corrosion, at thin metal sections as in bubble caps, etc., and in high velocity pumps, the economic balance between plant investment and maintenance can be good to excellent.

The hot patassium carbonate process is finding an important economic place in the sweetening of reasonably large quantities of natural gas containing in the range of 8% or more of acid gas. This type plant is developing a dependability that only years

of experience can demonstrate. The corrosion problems are not as severe as in the amine plant, and other operational problems are familiar to experienced operators of sour gas plants.

GIAMMARCO-VETROCOKE PROCESS

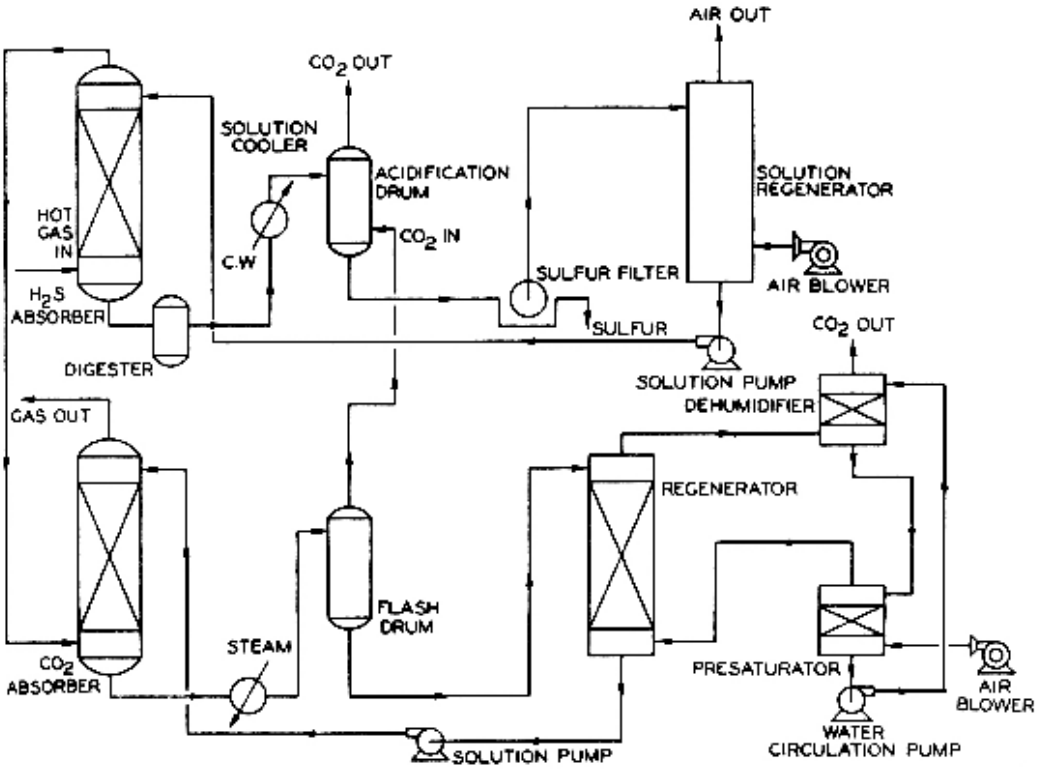
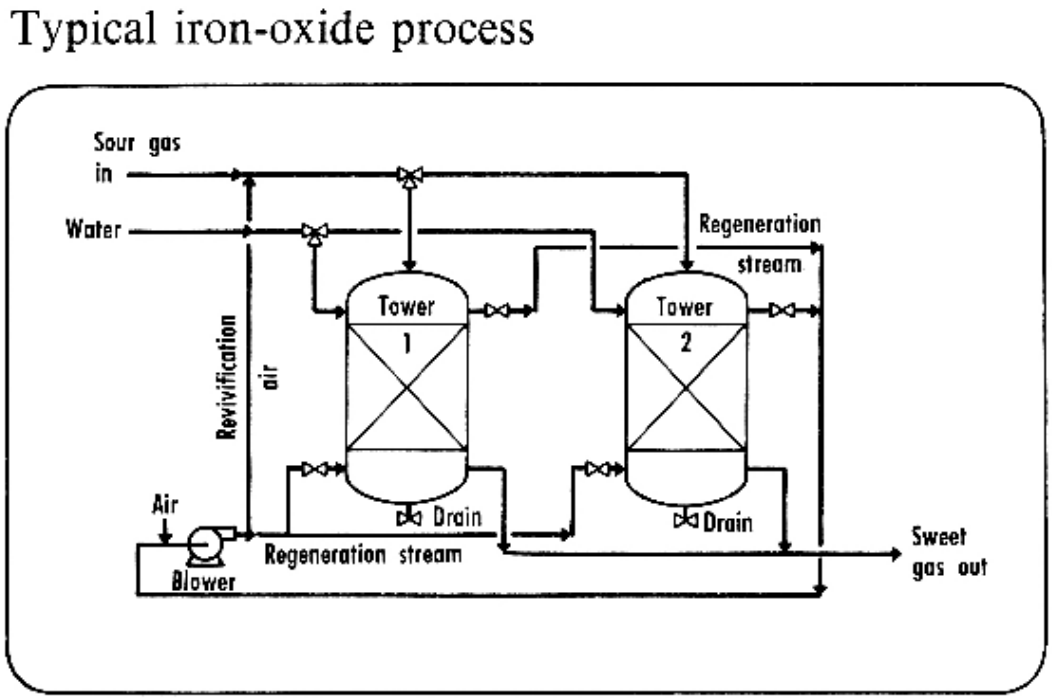


Figure shows the general flow for H_2S and CO_2 removal using air regeneration. Removal of CO_2 is accomplished in an alkaline solution containing inorganic additives such as arsenic oxide and/or selen-ious and tellurous acid. H_2S is removed by an alkaline solution of arsenites and arse-nates. Simultaneous absorption of H_2S and CO_2 can be obtained by proper selection of the proportion of arsenite to arsenate and pH of the solution.

At present this process is not considered applicable where the inlet H_2S content is greater than 1.5% or the total sulfur removed exceeds 15 tonnes per day.
Within their limitations, both the G-V and potassium carbonate processes offer economic advantage over the amine process. However, when the gas specification is 0.6 g/100 std m^3, an amine process is usually necessary to "clean up" the gas following these processes.

IRON SPONGE PROCESS



Flow Sheet For Iron Sponge Process.

Figure 10.11 shows this process, which is selective for H_2S only. It often offers advantage when the sulfur in the gas does not exceed 7-9 tonnes per day and the concentration does not exceed 2400 g/ 100 std m^3 [1000 grains H_2S per 100 scf] of gas.

This process consists of wood chips impregnated with varying amounts of hydrate iron oxide (Fe_2O_3). This reacts with the H_2S to form FeS_3, which may be regenerated with air. Continuous regeneration is possible by injecting a small stream of air into the feed-gas stream which converts the sulfide to the oxide and liberates the elemental sulfur. Regeneration is commonly done periodically, however, by flowing air through the bed. Regeneration is normally finished when the outlet oxygen concentration reaches 4-6% and the bed outlet temperature starts dropping. Each charge of sponge may be regenerated several times, but it gradually becomes less efficient and requires replacement.

The iron sponge process has the advantage of:

1. Providing complete removal of small to medium concentrations of hydrogen sulfide without removing carbon dioxide.
2. Requiring relatively small investment, for small to moderate gas volumes, compared with other processes.
3. Being equally effective at any operating pressure.
4. Being used to remove mercaptans or convert them to disulfides.

The disadvantages of the process are:

1. It is a batch process, requiring duplicate installation or flow interruption of processed gas.
2. It is prone to hydrate formation when operated at higher pressures and at temperatures in the hydrate-forming range.
3. It will effectually remove ethyl mercaptan that has been added for odorization.
4. Coating of the iron sponge with entrained oil or distillate will require more frequent change out of the sponge bed.

The wood shavings serve only as a carrier for the hydrated iron oxide, and as a packing to control gas distribution and pressure drop. Several grades, based on iron oxide content, are available. In traditional English units these have been 6.5, 9.0, 15.0 and 20.0 Ib per bushel of shavings. The equivalent metric units are 84, 116, 193 and 258 mg per cubic meter of shavings. In a properly packed vessel about a 20% shrinkage occurs due to compaction.

There are a few basic concepts for the design of the iron sponge vessels:

(1) The oxide bed should be at least 3 m deep to produce pressure drop sufficient to give proper distribution over the entire cross section area of the vessel. This is a good depth, Increased depth, unless supported on trays, etc., produces excessive pressure drop, due to compaction. Increased depth is worthy of consideration, however, as shell length is a relatively inexpensive manner of increasing bed life and minimizing the expense of change out of the charge.

(2) The vessel should be of such diameter as to limit sulfur deposition to a maximum of 20 450 mg/m^2[15 gr/sq.ft.] of cross-sectional area of bed per minute. This rule must consider the number of grains of sulfur that will be present under operating conditions that might be several atmospheres of pressure and thus several times the concentration of the standard test results. This rule becomes critically important when designing for high concentration of sulfur or for high operating pressures. The indicated loading should not be exceeded, and the lower it can be held, the more complete the chemical reaction between the sulfide and the iron oxide.

Lower loading will make for closer approach to the theoretical reaction of 0.56 kg of sulfur per kg of iron oxide present. A loading of 0.50-0.52 is common.

(3) The time of contact in the bed can be most easily figured by assuming a sponge bed volume equal to the actual bed depth, times the cross section area of the vessel, empty of sponge. Using again the volume of gas under actual operating conditions (preferably minimum operating pressures in case of variable operating pressures), calculate the resident time, or the time of contact of the per m^3 of sponge volume and a minimum of 50 m^3/h under any condition, or a minimum of 90 m^3/h where there is over 1200 mg/std m^3 [50 gr/200 scf] and no oxygen is present.

Excessive velocities will cause excessive pressure drop, abnormal compaction of the bed, and "break through" of sour gas at the bed outlet before the iron sponge has been completely chemically converted to either iron sulfide or free sulfur.

Minimum velocities below 50 $m^3/h/m^3$ bed can possibly cause gas to channel, or not to move at sufficient rate to reduce the gas film at the surface of the sponge to permit new foul gas to contact the active sponge.
Mass velocity in kg/m^2/min should not exceed 48 times the square root of the gas density in kg/m^3 for upflow or 96 times for downflow.

Ranges of flow rates between these maximum and minimum rates are a function of the oxygen content and operating temperature. Without oxygen present, the sulfiding reaction produces only 610 kJ/kg of hydrogen sulfide. If the operating temperature is held below 45°C, the essential water of crystallization will not be driven off and the adsorption process can be continued at design rate. However, if atmospheric oxygen is present in the gas from vacuum compression, or oxygen has been added in controlled ratio to reactivate the sponge simultaneously with the adsorption process, the heat of reaction can approach 5935 kJ/kg of hydrogen sulfide and cause permanent loss of the water of crystallization at operating temperatures above 45°C. If excess oxygen is permitted (i.e., over 0.4% pure oxygen plus 0.1% for each 1200 mg/std m^3 of H_2S, the loss of this water of crystallization can occur in the bed, if the temperatures are permitted to rise unchecked. The active bed should be held to a weight basis moisture content of 40%, plus or minus 15%. The original sponge is processed to a pH of 8 or above and should be held to that value by addition of soda ash as needed.

(4) Without oxygen present, the theoretical conversion rate of 0.56 kg of sulfur per kg of iron oxide can be approached under ideal conditions. Under ideal conditions of oxygen regeneration, either simultaneously or in batch

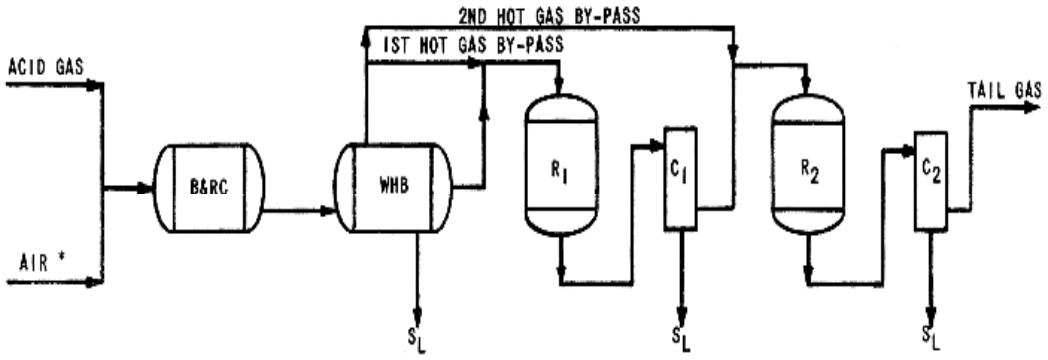
regeneration, ideal conversion rates of 2.5 kg of sulfur per kg of iron sulfide can be achieved. The life of a batch or a charge must be considered if the size of the bed, as indicated by Item 2 and/or Item 3 above, is so small as to. require frequent change out. The labor of changing out a batch can be a high percentage of the overall cost of operation. The cost of the sponge itself is not a variable item as related to kg of sulfur removed. If regeneration is not to be considered, then the 193 mg/m^3[15 Ib/bushel] sponge should be used. This weight of iron oxide will give maximum utilization before the void space is filled with iron sulfide to produce an excessive pressure drop. If air regeneration is to be used, the 116 mg/m^3[9 Ib/bushel] sponge will possibly give a better record of purification but might be blanked off by sulfur before it is utilized to the full extent of its potential reaction. For any given weight of iron oxide, the amount of sulfide that a given bed will remove can be rather closely calculated, except for blanking-off by excessive sulfur or by oil brought into the bed, which will seal off the iron oxide and prevent chemical reaction.

OTHER PROCESSES

The above are most commonly used for pipeline sweetening of natural gas. However, many other processes are available.

RECOVERED SULFUR PROCESSES

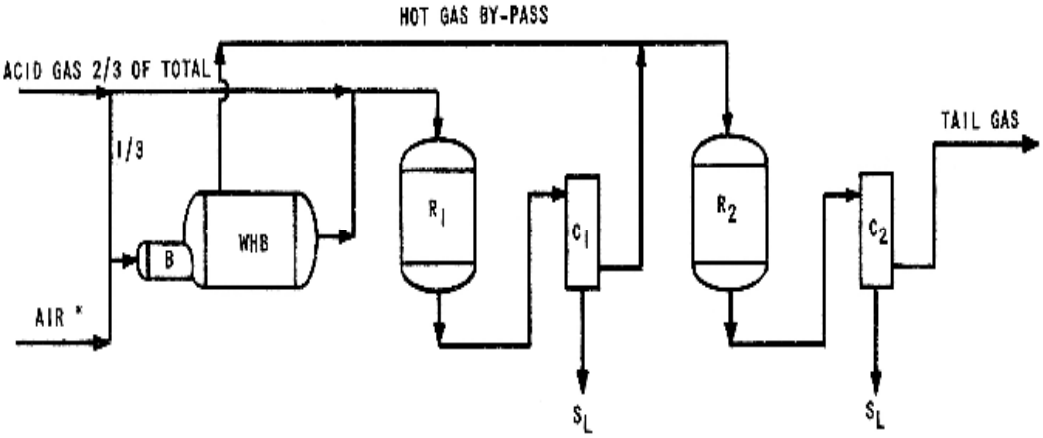
Current sulfur processes may be classified into several general categories. The most common historically has been the modified Claus process shown in Figure 10.12. It may use various modifications of the once-through process or the split-stream process. If the H_2S content of the acid gas is 20-25%, the split-stream process may be used to sustain a flame in the reactor furnace. Otherwise, the once-through process would be preferred, for it gives higher overall recovery and heat efficiency.



LEGEND

B	- BURNER	R_2	- SECOND CATALYTIC CONVERTER
RC	- REACTION CHAMBER	C_1	- FIRST CONDENSER
WHB	- WASTE HEAT BOILER	C_2	- SECOND CONDENSER
R_1	- FIRST CATALYTIC CONVERTER	S_L	- LIQUID SULFUR

Sufficient air is added to burn 1/3 of total H_2S to SO_2 and all Hydrocarbons to CO_2
Straight Through Flow Scheme



LEGEND

B	- BURNER	R_2	- SECOND CATALYTIC CONVERTER
WHB	- WASTE HEAT BOILER	C_1	- FIRST CONDENSER
R_1	- FIRST CATALYTIC CONVERTER	C_2	- SECOND CONDENSER
		S_L	- LIQUID SULFUR

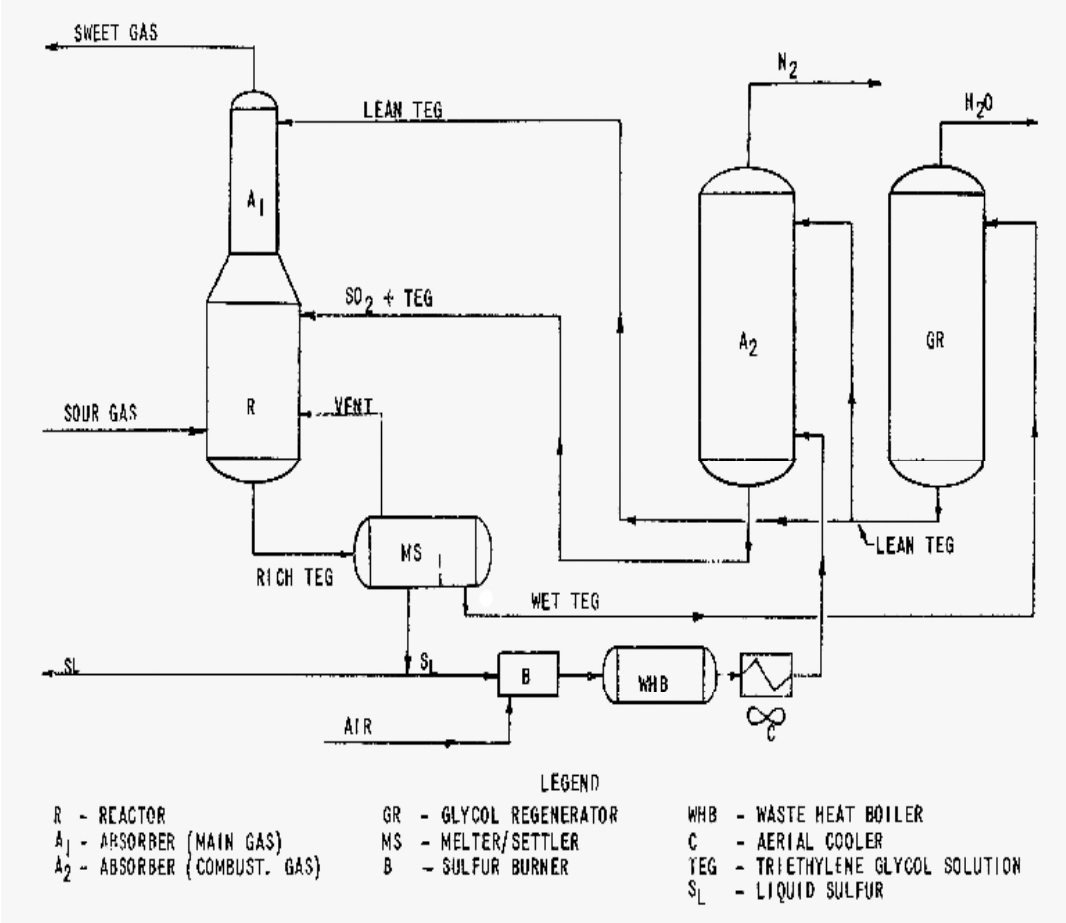
Sufficient air is added to burn all H_2S to SO_2
and all Hydrocarbons to CO_2 in 1/3 of acid gas
Straight Through Flow Scheme

Typical Flow Sheet for the Modified Claus Process

The Modified Claus process is normally used on the regeneration gas from an amine-type absorption plant.

A direct oxidation process has also been developed by Pan American Petroleum Corporation for gases containing about 2-15% H_2S in the acid gas stream.

Plants using a combination of absorption and air oxidation to produce sulfur are available The Giammarco-Vetrocoke system has been briefly described previously.
The Townsend Process is one of the new types of absorption processes using direct conversion in this process lean glycol dehydrates the gas. Glycol is used to absorb SO_2 formed by burning a portion of the sulfur production. The SO_2-rich glycol reacts with H_2S to form elemental sulfur it is in the pilot plant stage at present but shows some promise. A general flow sheet of this process is shown in Figure 10.13.



Flow Sheet of the Townsend process

MODULE 5:

Hydrocarbons Recovery

HYDROCARBONS RECOVERY

INTRODUCTION

Recovery of natural-gas liquids (NGL) from natural gas is quite common in natural gas processing, and can, at times assume great economic importance. Huge quantities of solution gas produced from crude oil used to be flared routinely. Phasing out this wasteful practice has required greatly increased NGL recovery.

Prior to 1984, the value of ethane as a raw material for petrochemicals was sufficiently high that its recovery was economically desirable. Figure 14-1 (Mehra, 1987) shows recent trends in sales gas and product value in the United States. The value of sales gas declined from about $3.50/ MMBtu in early 1984 to about $1.50/MMBtu in late 1986 (West-Texas intrastate prices). As a liquid, propane was worth sufficiently more than its value in sales gas to warrant investment in liquid recovery for the period 1984-85, but during 1986 there was very little economic incentive for propane recovery. During most of this period liquid was worth less than its value in sales gas.

One factor in this price reversal is that relatively-cheap naphtha, as well as n-butane, presently compete as raw materials for ethylene production. However, some ethylene plants cannot crack naphtha. Even though there is little economic incentive to recover ethane, petrochemical plant expansion is currently occurring in the United States, and this may lead to a better ethylene (and thus ethane) markets.

The value of n-butane also varies widely. Normal butane is currently used to control the vapor pressure of automotive gasoline. Butane is desirable for this purpose since it has a high vapor pressure of 52 psia and a road octane number of 91.7. The U. S. Environmental Protection Agency (EPA) has ruled that, starting in 1989, the vapor pressure of automotive gasoline must be reduced, which means less blending of n-butane. In the United States, refineries purchase essentially all current n-butane production from natural-gas processing plants for use as a blending stock. With this mandated reduction in vapor pressure, refineries will be potential sellers of n-butane instead of purchasers. At present, other uses for n-butane include isomerization to i-butane, feed stock in ethylene manufacture, and boiler fuel. In summary, the present outlook for natural-gas liquids sales in the United States is soft, although King (1988) indicates that prices have stabilized somewhat. Figure 14-2 reports recent NGL prices (Brewster, 1986-89) compared with natural gas prices (Oil & Gas J., 1986-89) on an energy-equivalent basis.

Because ethane is sometimes more valuable as NGL and at other times worth more as sales gas, NGL-recovery plants need to be versatile—capable of either producing or rejecting ethane liquid product.

As stated in Chapter 5, NGL recovery is not totally an economic matter. Production of gas that meets stringent sales specifications (e.g., HC dew point) may require NGL extraction regardless of economics. Thus, a good sales-gas market can compensate for a poor demand for NGL. Finally, it may be desirable to produce a gas that can be pipelined without hydrocarbon condensate formation, thus requiring NGL removal. This chapter discusses NGL recovery beginning with a brief review of the processing objectives (see Chapter 5 for more detail). The economic value of NGL is presented in quantitative terms to underscore the relative merit of selling them as liquids or as sales gas. Various liquid-recovery processes are reviewed briefly as an introduction to process selection. Then the more important liquid-recovery processes are discussed in greater detail, prefaced by remarks on modern How-sheet simulation. Handling and separation of the liquid products complete the chapter.

PROCESSING OBJECTIVES

The three alternatives of liquid-recovery processing are: to produce transportable gas, to meet sales-gas specifications, or to maximize liquid recovery (Odello, 1981).

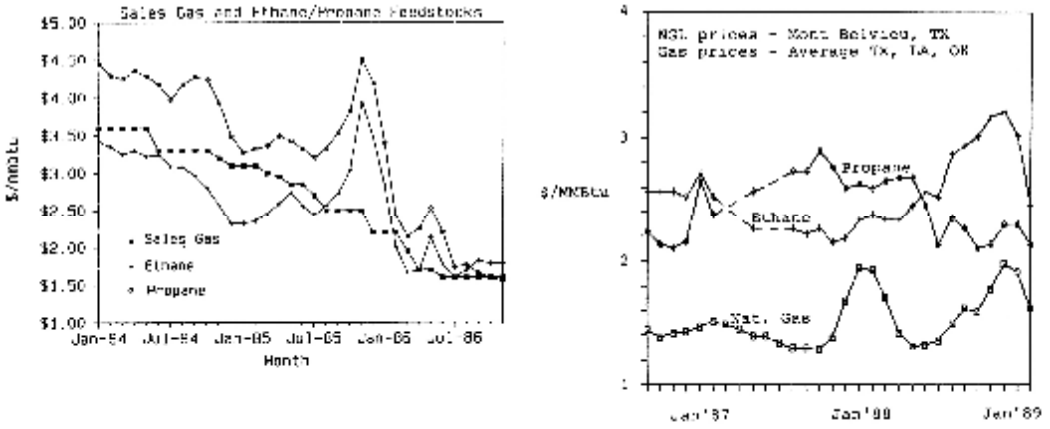
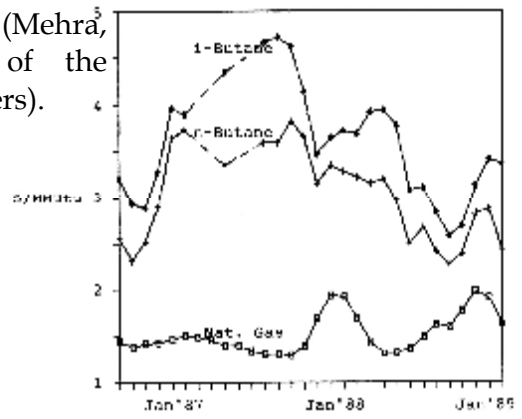


Figure 14-1. Energy equivalent prices (Mehra, 1987. Reproduced by permission of the American Institute of Chemical Engineers).



Transportable Gas

For remote locations such as offshore or the outback, the only alternative is to produce gas that can be pipelined without condensation. Condensation has two drawbacks. First, two-phase flow requires a larger pipe diameter than single-phase flow for the same pressure drop. Second, when the two-phase stream arrives at its destination, elaborate slug-catchers may be required to protect the equipment downstream (Huntley and Silvester, 1983).

There are two expensive alternatives: NGL recovery at the remote site or dense-fluid pipelining. However, depending on the economics of the particular case, the least expensive option for the remote location can be any one of these three options:

1. Recover only enough NGL to obtain single-phase flow.
2. Transport the gas in two-phase flow.
3. Transport the gas in dense-phase flow.

These options are explored further in Chapter 5. Note that minimum NGL extraction is a practical objective in some cases.

Sales Gas

Perhaps producing sales gas with acceptable specifications is the most common choice. Generally there is a minimum gross heating value (GHV) specification and possibly a HC dew-point requirement. If more valuable as liquids, NGL removal should be maximized while still satisfying the minimum heating value specification. If NGL is worth more in the gas phase, it may be desirable to retain them as gas, subject to the HC dew-point requirement.

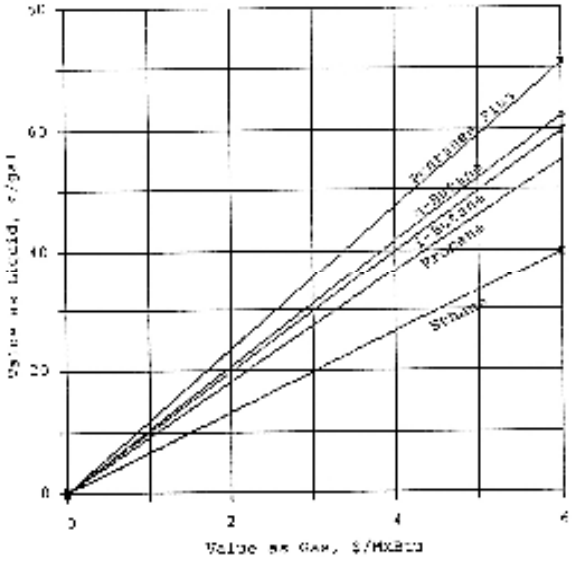


Figure 14-2. Relative price of gas and natural gas liquids.

Maximum NGL Recovery

A normal heating-value specification of about 1000 Btu/ scf can be met with methane alone as is shown below.

Hydrocarbon	GHV(Btu/scf)
Methane	1010
Ethane	1770
Propane	2516

However, many natural gas streams contain N_2 and/or CO_2, and these incombustible gases can require presence of ethane to provide the required heating value. If the heavier hydrocarbons are more valuable as liquids, essentially complete liquefaction of propane and heavier hydrocarbons and partial ethane recovery is desirable.

A second case for maximum liquid recovery is the cycling of natural gas in a condensate reservoir. Now all or a portion of the produced gas is reinjected to keep the reservoir pressure above the gas dew point and thus maximize ultimate NGL recovery. If the reservoir pressure is allowed to fall into the two-phase region, valuable liquids are condensed and will not be recovered.

VALUE OF LIQUEFIABLE COMPONENTS OF NATURAL GAS

Potentially, ethane and heavier HC components (C2+) can be liquefied. At one time, ethane was not considered a valuable liquefiable component, mainly because no economic method existed for high-yield liquid recovery. The advent of the expander plant, coupled with high demand for ethane to manufacture ethylene, motivated high ethane recovery.

Relative liquid- and gas-phase values of hydrocarbons are now illustrated for propane. In sales gas, propane is worth the contract price of the gas, assuming it can be left in the gas and sold for its GHV. Suppose gas is worth \$3.00/MMBtu. The heating value of propane is 2516.1 Btu/scf (GPSA, 1987). Then

1000000 Btu x 1 scf/2516.1 Btu = 397.4 scf propane

If this propane is liquefied the amount of liquid recovered at 60°F is

397.4 scf x 1 gal/36.375 scf = 10.9 gal propane

where 36.375 scf/gal is read from Appendix 6 (Table A6-1) or Figure 23-2 (GPSA, 1987). The equivalent value of this propane as a liquid is, therefore,

\$3.00/10.9 gal = \$0.275/gal

when gas is worth \$3.00/MMBtu.

If liquid propane can be sold for more than $ 0.275/gal plus the cost of liquefaction, there is an economic incentive for propane liquifaction when gas is worth $3.00/MMbtu. Figure 14—3 shows the equivalence for NGL components as a function of gas price.

A similar argument can sometimes be made for "spiking'' the condensate recovered from an associated gas back into the crude oil. Crude spiking not only increases the total barrels of oil but also raises the API gravity, which may increase the sales price per barrel. Again, crude value is important; $0.275/gal converts to $11.55/bbl. The crude

Figure 14-3. Energy equivalent value of natural gas liquids.

would have to be worth more than $11.55/bbl to make the spiking economical.
As stated above, the extent of condensate removal may be limited by the sales-gas GHV specification, particularly if appreciable N_2 and/or CO_2 are present. Figure 14-4 (Ewan et al., 1975) shows how ethane recovery is limited by inert gas content. The graph is approximate, but illustrates the principle involved.

LIQUID RECOVERY PROCESSES

Figure 14—5 shows the phase diagram for a natural gas, either a gas-well or associated gas. Obviously any cooling will induce condensation and yield NGL. The higher the pressure, the more condensation, other factors being equal. Another NGL recovery technique is the use of so-called mass-transfer agents (MTA); two MTA processes are described later.

The basic NGL liquefaction processes are now described and related to Fig. 14—5, where possible. Additional details for several processes are given later.

Refrigeration

Refrigeration, shown in Figure 14-6a, is the simplest and most direct process for NGL recovery. External or mechanical refrigeration is supplied by a vapor-compression cycle that typically uses propane as the refrigerant or working fluid. Mechanical refrigeration is described in detail below.

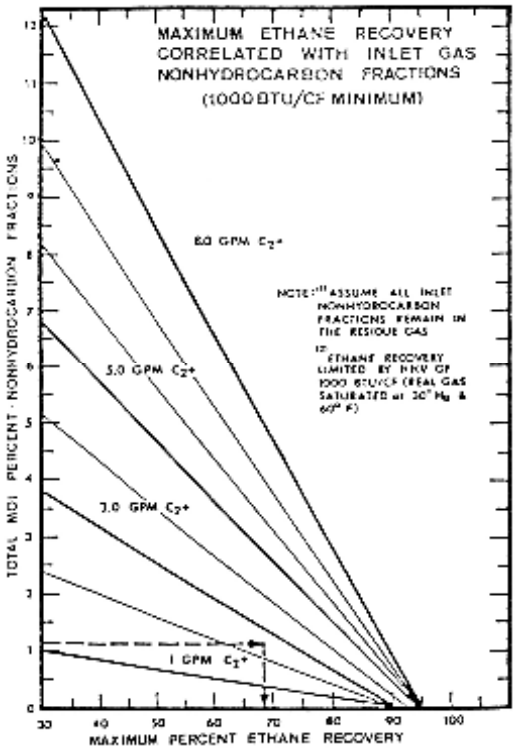


Figure 14-4. Maximum ethane recovery correlated with inlet gas nonhydrocarbon fractions (Ewan et al., 1975).

In Figure 14-6a, the gas-to-gas heat exchanger recovers additional refrigeration by passing the gas leaving the cold separator countercurrent to the warm inlet gas. The temperature of the cold gas stream leaving this exchanger "approaches" that of the warm inlet gas. Economically, this approach can be as close as 5°F or slightly less.

The chiller in Figure 14—6a is typically a shell-and-tube, kettle-type unit. The process gas flows inside the tubes and gives up its energy to the liquid refrigerant surrounding the tubes. The refrigerant (often C3) boils off and leaves the chiller vapor space essentially as a saturated vapor.

The refrigeration process is shown as line ABC in Figure 14-5. From A to B indicates gas-to-gas exchange; from B to C, chilling. Gas-to-gas exchange is very common in NGL recovery processes. Figure 14-6a is deceptively "simple," in that the condensed liquid must be fed to a stabilizer (or fractionation column) to meet typical

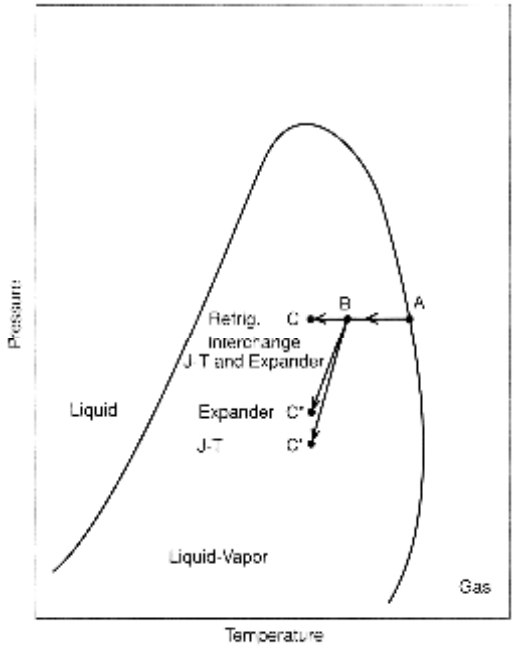


Figure 14–5. Phase-diagram paths for NGL recovery.

Figure 14-5. Phase-diagram paths for NGL recovery.

specifications such as vapor pressure or composition (%C2 if recovering C3 +).

Ethylene glycol is often injected at the inlet of the gas-to-gas exchanger and/or chiller to prevent hydrate formation, or freeze-up, in these exchangers. Freeze-up will partially block exchanger tubes, thus increasing pressure drop and decreasing heat exchange. The weak glycol solution, containing condensed water, is separated in the cold separator, reconcentrated, and recycled. Glycol injection is described in detail in Chapter 6.

Joule-Thomson Expansion (LTS)

Figure 14-6b shows the Joule-Thomson (JT), or low-temperature separation (LTS), process. The inlet gas passes first through the gas-to-gas exchanger, thence to an expansion or "choke" valve. The expansion is essentially a constant enthalpy process. Nonideal behavior of the inlet gas causes the temperature to fall with the pressure reduction, as shown by line ABC' in Figure 14-5. The temperature change depends primarily on the pressure drop. The JT process is a "self-refrigeration" process, as opposed to "external refrigeration." Again, the condensed liquids must be fractionated to meet vapor pressure and composition specifications.

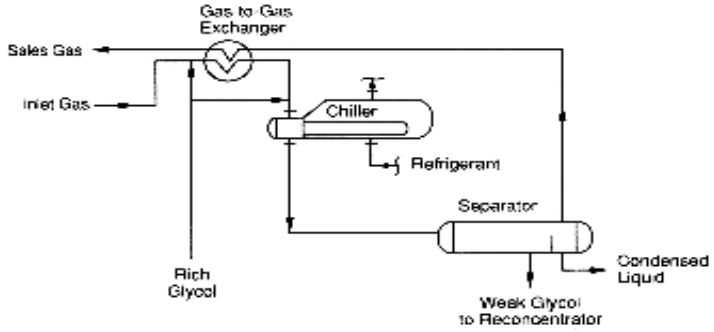


Figure 14-6a. Refrigeration.

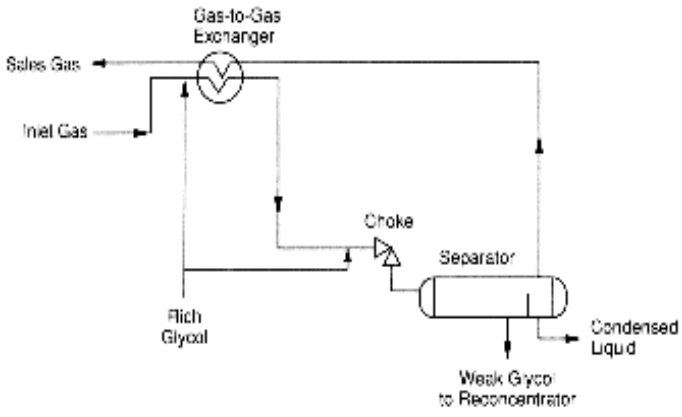


Figure 14-6b. Low-temperature separation.

Generally, sales gas must be recompressed to sales pipeline pressure if it has been expanded to a lower pressure. Thus this JT process is most favored when the wellhead gas is produced at a very high pressure and can be expanded to sales-line pressure with no recompression. If the gas must be recompressed, the JT process is penalized by the recompression horsepower requirement.

Expansion Turbine

In the so-called cryogenic or expander plant, the chiller or J-T valve is replaced by an expansion turbine. As the entering gas expands it supplies work to the turbine shaft, thus reducing the gas enthalpy. This decrease in enthalpy causes a much larger temperature drop than that found in the simple J-T (constant enthalpy) process. The expansion process is indicated as line ABC" in Figure 14—5. Figure 14-7 shows an expander-plant flowsheet that features two gas-to-gas exchangers and a C3 chiller. Feed gas is cooled first in the high-temperature, gas-togas exchanger (A) and second in the propane chiller (B). Condensed liquids then are removed in a separator (C).

Gas from this separator is cooled further in the low-temperature gas-to-gas exchanger (D) and fed into a second cold separator (E). Gas from the cold separator expands through the expansion turbine (F) to the pressure at the top of the demethanizer (G), which varies from 100-450 psia. During the expansion, condensate is formed but the turbine is designed to withstand this condensation. Typically 6 to 8% of the feed gas is condensed in the cold separator which is usually at -30 to -60°F. The expander lowers the pressure from the inlet gas value, say 600 to 900 psia, to the demethanizer pressure of 100 to 450 psia. Typical inlet-gas temperatures to the demethanizer are — 130 to — 150°F, sufficiently low that a great deal of the ethane is liquefied. Expander plants typically condense 8 to 12% of the feed gas.

The demethanizer tower stabilizes the raw liquid product by reducing the methane content to a suitably low level. The molar ratio of methane to ethane in the bottom product is typically 0.02 to 0.03. The bottom product temperature is often below ambient, so that feed gas may be used as the heat-transfer medium for the reboiler. This provides additional refrigeration to the feed and yields higher ethane recovery, typically 80%. Additional refrigeration also can be obtained by using the feed gas for side-reboiling heat in the demethanizer.

If the feed gas is lean, say < 2.5 to 3 gal NGL/ Mscf (GPM), auto refrigeration (gas-to-gas heat exchange) usually suffices. For moderately rich feeds, say >3 GPM, mechanical refrigeration should be considered to obtain high ethane recovery most economically—shown in dotted lines in Figure 14-7. In fact, mechanical refrigeration alone is probably the most attractive NGL recovery process for very rich (> 10 GPM) feed gases.

In expander plants, the low temperatures make material selection very important (McKee, 1986). Carbon steel is suitable down to — 20°F but becomes brittle at lower temperatures. Charpy-impact-tested carbon steel can be used as low as -SOT. Between -50 and -150°F, 3.5% Ni steel is used, and stainless steel below — 150°F.

The extremely low temperatures also require a very low water content in the inlet gas to prevent hydrate formation throughout the cryogenic process. Solid-desiccant dehydration is required (see Chapter 9 for details).

If the feed gas contains more than about 0.5-1.0 mol percent CO_2, solid CO_2 may form in the expander outlet gas or in the demethanizer near the top. This may dictate CO_2 removal from feed when the heating-value specification would not dictate such.
Approximately 98% of the energy lost in the expander is recovered in a brake compressor that partially recompresses the residue gas. Additional recompression is normally required to restore the gas to sales-line pressure.

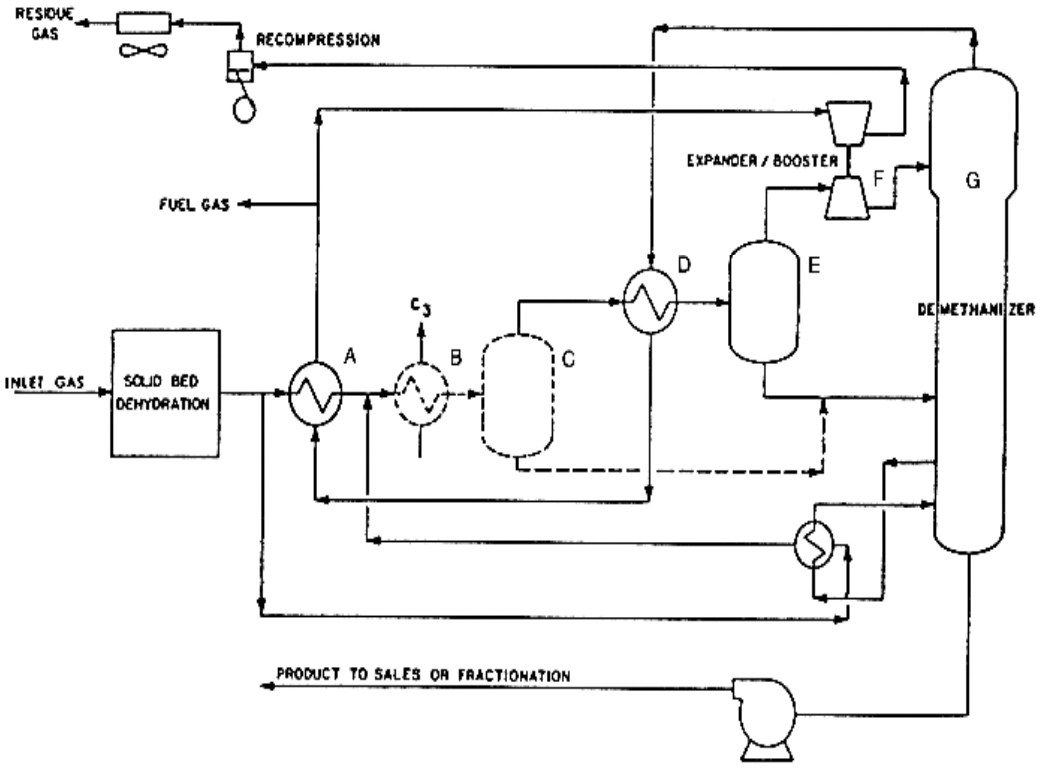


Figure 14-7. Process flow diagram turbo-expander gas liquids extraction plant (Ewan ef a/., GPA Proceedings, 1975).

Oil Absorption

Absorber oil can be used as a mass-transfer agent (MTA) to recover liquids from natural gas with only a 5-10 psi pressure drop in the absorption tower (Fig. 14-8). If the low-pressure feed gas must be compressed to sales-gas line pressure, the tower pressure can be selected to optimize the absorption process. If, on the other hand, the feed gas is available at sales-line pressure, then absorption is carried out at pipeline pressure, thus requiring very little, if any, recompression. With no recompression, 30 to 50 psi are required to flow the feed through the plant. A condensate recovery plant that operates on a high-pressure pipeline and returns the residue gas at pipeline pressure is sometimes called a "straddle" plant.

The trade-off for low pressure drop is that a complex recovery process is required to separate the absorbed NGL from the absorber oil and to remove dissolved methane and/or ethane from the absorber oil (Fig. 14-8). The process has high fuel requirement, even with efficient energy recovery. Absorption oil plants still remain in operation, but many have been replaced by simpler, more-efficient expander plants.

The absorber oil and the inlet rich gas are almost always chilled to increase NGL recovery. Typically such "cold oil" plants operate with temperatures of +30 to -30°F for the streams entering the absorber.

As noted above, lean-oil recovery starts with the removal of dissolved methane. The demethanizer (DEC1ER) tower is reboiled for methane stripping, but valuable liquefied ethane and propane are vaporized also, thus requiring a cold-oil reabsorber section in the top of the tower. In cold-oil plants that do not feature ethane recovery, the demethanizer tower becomes a deethanizer tower.

Second, the absorbed condensate is separated from the lean oil in the still. The raw condensate passes overhead. The hot lean oil from the still bottom is returned through the demethanizer reboiler(s) for energy conservation, then cooled and chilled.

Absorption oil is a clear HC liquid similar to gasoline or kerosine, but with a narrower distillation range. Absorption oil has an average molecular weight (MW) of 100 to 140, depending on the temperature level in the absorber column. For -30°F inlet temperature, a 100 MW oil is satisfactory. For + 30°F inlet temperature, 140 MW is appropriate. Table 14—1 gives a typical analysis for a cold-oil-plant lean oil.

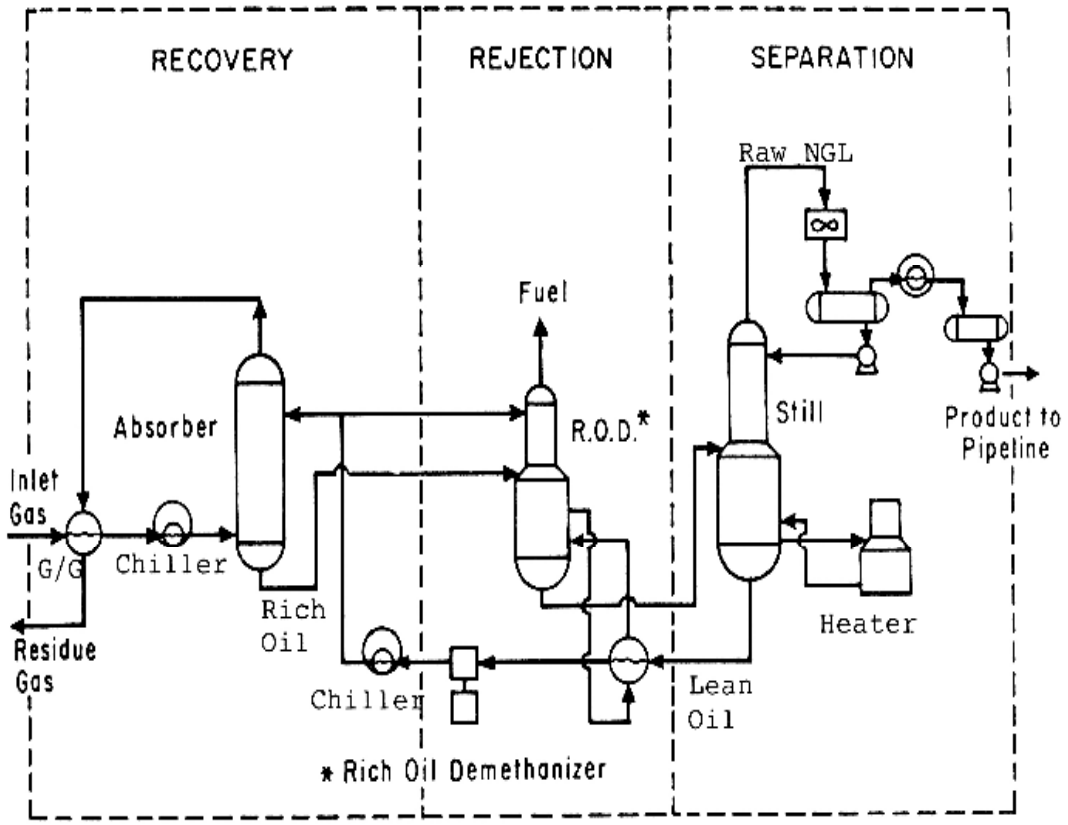


Figure 14-8. Oil absorption plant (PETEX, 1974. Copyright © 1974 by the University of Texas at Austin).

Table 14-1 Properties of a Typical Absorber Lean Oil

Components	Analysis Mole %	Average Carbon Number of Cut	ASTM Distillation	
			Vol. %	Temp °F
Propane	0.0061	3	IBP	242
i-Butane	0.0092	4	5	270
n- Butane	0.013	4	10	283
i-Pentane	0.023	5	20	294
n-Pentane	0.017	5	30	300
Hexanes	0.45	5.92	40	306
Heptanes	2.09	6.91	50	312
Octanes	10.20	7.71	60	319
Nonanes	32.96	8.71	70	328
Decanes	34.00	9.64	80	340
Undecanes Plus	20.23	----	90	361
			95	386
			EP	435
			Recov.	98.6
			Loss	1.4
PONA Analysis:	Paraffins 52.2	Sp. Gr. @ 60°F/60°F	0.7703	
	Olefins 0.3	Molecular Weight	130	
	Naphthenes 35.6			
	Aromatics 11.9			

Data from a report by P-V-T, Inc., Houston, TX

Solid-Bed Adsorption

Solid desiccants, such as silica or molecular sieves, and activated charcoal can be used as an MTA for condensate recovery (Fig. 14—9). The process is continuous with respect to the gas but cyclical with respect to the adsorbent bed because the latter must be regenerated when it becomes saturated with condensate. Regeneration is accomplished by passing heated recycle or bypass gas through the bed. The condensate is recovered from the regeneration gas by cooling, condensation, and phase separation. A typical cycle time for the process is two or three hours. For this reason the process is sometimes referred to as "quick cycle" or "short cycle" adsorption. The units are also called HRU (hydrocarbon recovery units). See Chapter 9 for more details on solid-desiccant adsorption.

Short cycle units are energy intensive because of the regeneration process. The adsorption beds are heavy and expensive. This process option is not often used but may be considered in special applications, such as HC dew-point control in remote locations.

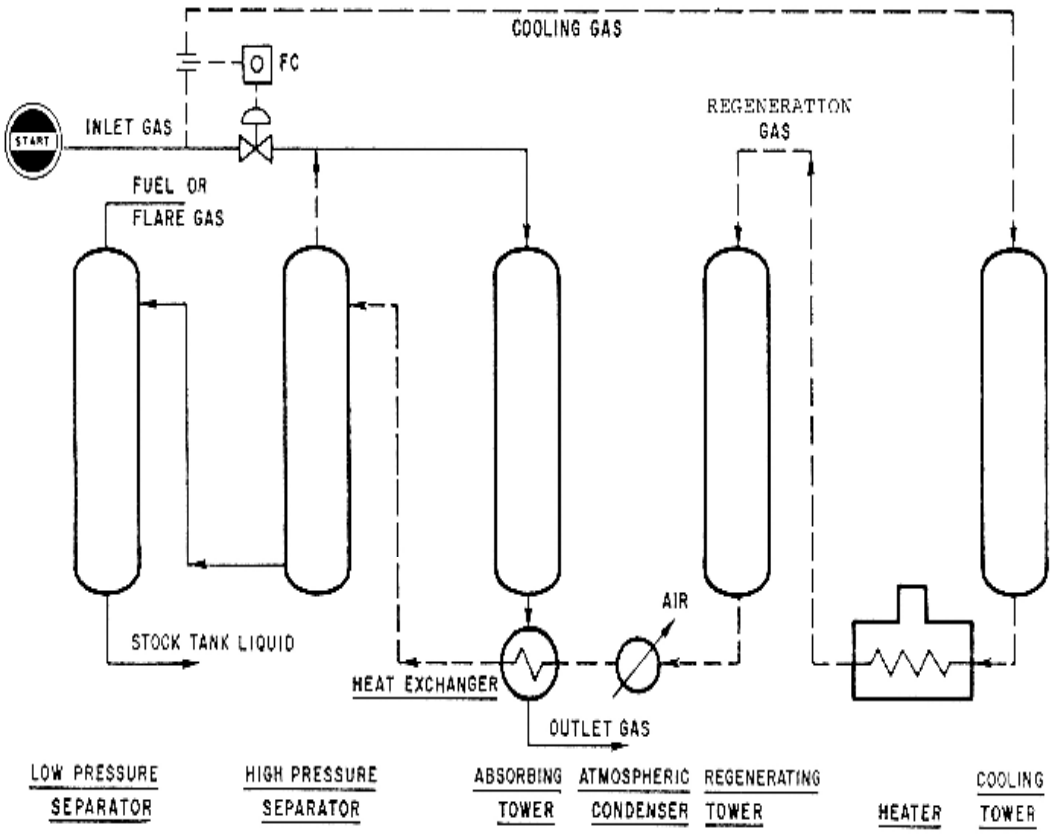


Figure 14-9. Solid-desiccant adsorption (Ballard, 1965).

PROCESS SELECTION

Quick generalizations regarding selection of a condensate recovery process are difficult. In some cases, an economic comparison between viable alternatives will be required, however, a few guidelines can be offered.

If the NGL content of the feed gas is low, the expander process will probably be the method of choice. For gases very rich in NGL, simple refrigeration is probably the best choice, while oil absorption or expansion are usually not satisfactory.

If the inlet gas pressure is very high, low temperature separation (LTS) may be attractive; this process has been used in recent years, both on and off shore. It is important that the reservoir pressure remain high for the intended life of the plant. Low inlet gas pressure favors an expander plant or straight refrigeration (if the gas is very rich).

Very low gas rates may justify only a very simple process, such as an automatically-operated JT unit. Large flow rates justify a more complex plant with more complex controls and more operating personnel.

Especially if few in number, remote wells may dictate simple operation and processing, such as LTS. A large number of wells may justify a central processing facility with more complex processing. Offshore locations usually demand simple processing due to expensive platform area and low weight requirements.

In summary, unique factors influence the process choice for each application. More and more expander plants are being used due to their simplicity and low operating cost. Such plants also can be adapted to ethane rejection, making them more versatile and responsive to product prices.

FLOWSHEET SIMULATION

The design of gas processing plants requires many repetitive calculations. In the past, design calculations were done entirely by hand, so that charts and tables of thermodynamic and other properties were indispensible tools. Beginning about 1960, many calculations were programmed for mainframe digital computers, which greatly improved the ability of designers to do their work efficiently and accurately. Many problems could be solved that previously could not be done at all by hand.

The introduction of hand-held calculators around 1970 increased hand computing power a great deal, particularly after programmable calculators were introduced. Microcomputers have totally changed design; almost all the required calculations can now be done at the desk using a stand-alone microcomputer. Recent developments

in equation-of-state computations and in available simulator packages underlie these changes and are summarized below.

Equations of State

In recent years, significant breakthroughs have been made in predicting pressure-volume-temperature relations and other thermodynamic properties using equations of state (EOS). The pioneering work of Georgio Soave (1972) extended use of the Redlich-Kwong EOS (1949) and the P'tizeretal. acentric factor (1961) to individual compounds. Special mixing rules permit prediction of not only accurate mixture enthalpies and entropies but also vapor-liquid equilibrium K-values. Shortly thereafter similar treatments were developed by other investigators, such as Peng and Robinson (1976). The latter treatment allows predictions for mixtures containing H_2S and CO_2 and was later extended to treat H_20 and hydrates.

Explanation of these EOS and their use are beyond the present scope. However, the basic procedure is the same as that for hand calculations by corresponding-states methods. Ideal-gas heat capacities are used to establish ideal-gas enthalpies and entropies. Then EOS are used to evaluate enthalpy and entropy deviations from ideal gas behavior for both real gases and liquids, as well as vapor-liquid equilibrium K-values. In turn, the K-values are used to establish the number and composition of phases present by standard flash calculations.

These EOS-prediction calculations are sufficiently complex that they would never be applied without digital computers. The advent of microcomputers made such calculations practical for any user.

The more important EOS methods are listed in Table 14—2 to provide guidelines for the inexperienced user. One need not be a theoretical thermodynamicist to use the methods, but one should know when the various methods are applicable.

Software Packages

Commercial software packages utilizing EOS to predict volumetric, thermodynamic, and phase-equilibrium properties incorporate the methods into flow-sheet simulation programs. Several methods were developed for the large, main-frame computer long before the microcomputers were available, e.g., Monsanto's FLOWTRAN, Chemshare's DESIGN 2000, and Simulation Science's PROCESS. Many operating companies developed their own in-house simulator packages.

These packages require input data files that are tedious to prepare; further, they are executed in batch mode on main-frame computers which makes turn-around slow. The programs are difficult for the casual user; experience is necessary for efficient use.

Table 14-2
Equations of State for Thermodynamic Properties of Mixtures in Gas-Processing Calculations

Equation of State	Comments
Grayson-Streed version of Chao-Seader	Used for refinery simulation, especially when H_2 present. Good to moderate pressures, temperatures ambient and above. Fastest method.
Soave-Redlich-Kwong (SRK)	Used for light hydrocarbons and natural gas processing to high P and low T (say, 2000 psia and -170°F). Will handle moderate amounts of CO_2, H_2S, and H_2.
Peng-Robinson w/Ng-Robinson (PR)	Similar to SRK, perhaps better in crit. region and for CO_2, and H_2S. Predicts gas-hydrate and CO_2,-frost points.
Benedict-Webb-Rubin-Starling (BWRS)	Superior to SRK and PR where applicable. Components limited to C1-C11, C_2=, N_2, CO_2, and H_2S. Much slower than SRK and PR, better liquid densities.

Table 14-3
Flow-Sheet Simulator Programs for IBM-AT Compatibles

Program Name	Source
ASPEN	JSD Simulation Service Co. Denver, CO 80222
CHEMCAD	COADE 952 Echo Lane, Suite 450 Houston, TX 77024
DESIGN II	ChemShare P.O. Box 1885 Houston, TX 77251-1885
HYSIM	Hyprotech Ltd. 400, 119-14SINW Calgary, Alberta, CANADA T2N 1Z6
PRO/II	Simulation Sciences Inc. 1051 W. Bastanchury Fuller-ton, CA 92633

The small, powerful microcomputers allowed development of flow-sheet simulator packages for the personal computer. Several programs have been converted to "friendly" or interactive form for data input, thus greatly simplifying their use. Today the computational power available to process engineers is enormous. Several of the better-known packages are listed in Table 14-3.

ADDITIONAL PROCESS INFORMATION

Additional details for several of the NGL recovery methods are discussed in the following sections.

Refrigeration Cycle

The basic or simple refrigeration cycle is described first. Improvements on the simple cycle and the effect of refrigerant purity on cycle horsepower are then described. Finally, mixed refrigerant systems are summarized.

Simple Cycle. The vapor-compression refrigeration cycle, or mechanical refrigeration process, is fundamental not only to the refrigeration process for condensate recovery but to other processes that use refrigeration as an integral part—such as expander plants and cold-oil absorption plants.

Pressure-enthalpy diagrams help illustrate refrigeration cycles and preliminary design calculations. Since the working fluid is usually essentially pure propane, a log P-h diagram similar to Figure 24-26 of the GPS A Engineering Data Book (1987) is convenient. Simple cycle calculations are shown in Figure 14-10.

The liquid leaving the condenser (point 1 in Fig. 14-10) is taken to be saturated at 125°F, and this corresponds to approximately 260 psia vapor pressure. The first step in the cycle is expansion of this liquid to the vapor pressure of the propane in the chiller where the temperature is specified (-10°F in the present case). This corresponds to about 32 psia in the shell side of the chiller. The JT expansion through the valve (to point 2) is essentially isenthalpic (constant enthalpy)—hence the vertical line on the log P-h diagram.

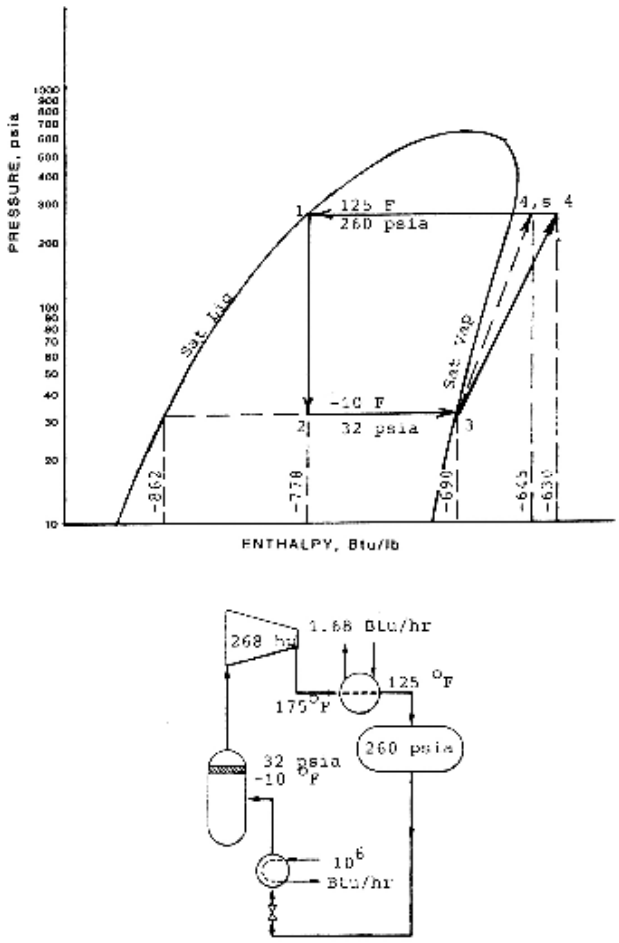


Figure 14-10. Propane refrigeration simple cycle.

During expansion, some liquid vaporizes, thus lowering the refrigerant temperature to the desired chiller value. The fraction vaporized is not essential to the calculation of the cycle but is easily obtainable by enthalpy balance.

$$mh_1 = mxh_{2,\text{sat vap}} + m\,(1 - x)\,h_{2,\text{sat liq}}$$

or

$$x = (h_1 - h_{2,\text{ sat liq}})/(h_{2\text{ sat vap}} - h_{2,\text{ sat liq}})$$

where m = refrigerant circulation rate

h = enthalpy

x = quality, mass fraction vapor

For the enthalpy values shown in Figure 14-10, the value of x is

$$x = (-778 + 862) / (-690 + 862) = 0.488$$

The remaining liquid fraction is available to be boiled and thus absorb process heat and so provide the desired refrigeration. This vaporization occurs at constant pressure and results in a saturated-vapor exit stream (point 3). The enthalpy change for this step represents the available chiller duty per each pound of refrigerant circulated.

$$Q_{ch} = h_3 - h_2 = -690 + 778 = 88 \text{ Btu/lb}$$

If the total duty, qch, is 1 MMBtu/hr, then the circulation rate is:

$$m = qch / qch = 10^6 \text{ Btu/hr } 4 \div 88 \text{ Btu/lb} = 113631\text{b/hr}$$

At the next step, vapor (point 3) is compressed back to the inlet condenser pressure (point 4), which is taken to be the outlet pressure plus a typical allowance for pressure drop in the condenser (5 psi). Compression is calculated in two steps. First, an isentropic path is followed to the final pressure (point 4,s). This theoretical, isentropic work of compression, W_{is}, equals the change in enthalpy.

$$- W_{is} = h_{4,s} - h_3 = - 645 + 690 = 45 \text{ Btu/lb}$$

This theoretical, isentropic work is now divided by an isentropic, or adiabatic, efficiency to estimate the actual work, W_{act}, done by the compressor.

$$-W_{act} = 45/0.75 = 60 \text{ Btu/lb}$$

(The 0.75 efficiency is a typical value for a centrifugal compressor. A value of 0.85 might be typical for a reciprocating compressor.) The actual final enthalpy, h_4, is calculated by assuming that the actual compression is adiabatic—all the work done on the gas ends up as enthalpy.

$$h_4 = -W_{act} + h_3 = 60 - 690 = -630 \text{ Btu/lb}$$

The actual temperature of the gas leaving the compressor is obtained by interpolation of the diagram at the final pressure and enthalpy and is approximately 175°F.

The gas (or fluid) horsepower is the work per Ib multiplied by the circulation rate, divided by an appropriate conversion factor, in this case 2544 Btu/hp-hr.

$$m (-W_{act}) = (11363)(60)/2544 = 268 \text{ HP}$$

The brake horsepower (input to the compressor shaft) is greater than the gas horsepower. For centrifugal compressors a quick estimate of the bearing and seal losses is simply estimated and added to the gas horsepower to obtain the brake horsepower. For reciprocating compressors it is customary to divide the gas horsepower by a mechanical efficiency to obtain the brake horsepower. See Chapter 10 for more information.

The final step in the cycle is the cooling and condensing of the compressed gas, shown by the approximately horizontal line (5 psi pressure drop) from point 4 to point 1. The condenser duty is

$$q_{cond} = m (h_1 - h_4) = (11363)(- 778 + 630)$$
$$= -1.68 \text{ MMBtu/hr}$$

Economizer Cycle. The efficiency of the simple cycle can be improved by introducing a flash economizer (Fig. 14-11). As before, start with the saturated liquid from the condenser, but flash it to an intermediate pressure rather than all the way down to the chiller pressure. A portion of the recirculating refrigerant, m_1, vaporizes in the economizer drum and is separated from the remaining liquid, m_2. Fraser (1984) found that the optimum economizer pressure is as follows.

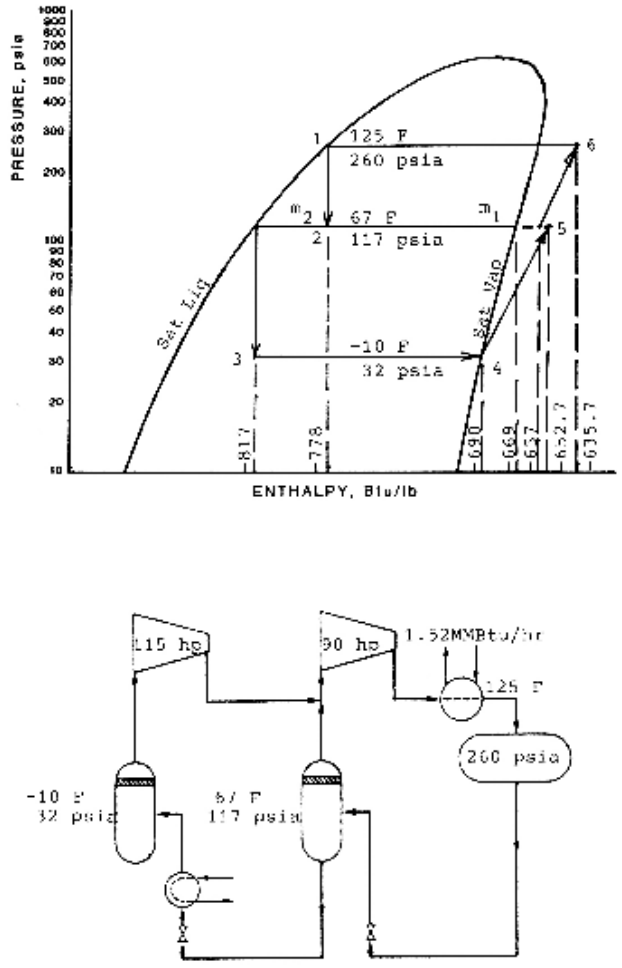


Figure 14-11. Propane refrigeration economizer cycle.

$$P_{econ} = P_{ch} (P_{cond,out}/P_{ch})^{0.614}$$
$$= 32 (265/32)^{0.614} = 117 \text{ psia}$$

Fraser points out that operation at pressures differing slightly from the optimum does not change the total horsepower greatly. The enthalpy change across the valve is zero and the enthalpy balance gives

$m_1 = m (h_1 - h_{2,sat\ liq}) / (h_{2,sat\ vap} - h_{2,\ sat\ liq})$
$= m (-778 + 817) / (-669 + 817)$
$= 0.264\ m$

where

m_1 = vapor flow from economizer drum
m_2 = liquid flow from economizer drum
m = liquid flow from condenser accumulator

But $m = m_1 + m_2$
So $m_1 = 0.264 (m_1 + m_2)$
Or $m_1 = 0.359\ m_2$

Liquid m_2, saturated at the intermediate pressure, then is flashed to the chiller, producing proportionately less vapor than in the simple cycle. Rate m_2 is

$m_2 = q_{ch} / (h_4 - h_3)$
$= 1 \bullet 10^6$ Btu/hr / (-690 + 817) Btu/lb
$= 7874$ Ib/hr

Thus

$m_1 = 0.359 (7874) = 2827$ Ib/hr
$m = 2827 + 7874 = 10701$ Ib/hr

The vapor from the chiller is compressed to the economizer pressure. As before, this compression work cycle is found in two steps.

$-W_{is}$ = -662 + 690 = 28 Btu/lb
$-W_{act}$ = 28 / 0.75 = 37.3 Btu/lb
$m_2 (-W_{act})$ = (7874) (37.3)/2544 = 115 HP
h_5 = -690 + 37.3 = -652.7 Btu/lb

The chiller vapor then is combined with the economizer vapor. The enthalpy of the mixture is found by enthalpy balance for this adiabatic mixing process.

$m_1 h_{2\ sat\ vap} + m_2 h_5 = m h_{5,m}$
$(2827) (-669) + (7874) (-652.7) = 10701\ h_{5,m}$
$h_{5,m} = -657$ Btu/lb

The combined vapor stream is now compressed to the condenser inlet pressure. The work is:

$-W_{is}$ = -657 + 641 = 16 Btu/lb
$-W_{act}$ = 16/0.75 = 21.3 Btu/lb
$m(-W_{act})$ = (10701) (21.3)/2544 = 90 HP
Total HP = 115 + 90 = 205
h_6 = -657 + 21.3 = -635.7 Btu/lb

Finally, the condenser duty is calculated.

$$q_{cond} = (10701)\ (-778 + 635.7) = -1.52 \text{ MMBtu/hr}$$

Considerable savings in circulation rate and in horsepower is achieved (205 vs. 268 hp). The importance of the horsepower savings lies primarily in the fuel savings for the compressor driver. For a reciprocating compressor there might be no reduction in frame size and virtually the same investment cost. In spite of the potential savings for the economizer cycle, the simple cycle is used a great deal in industry.

The GPS A Engineering Data Book (1987) has charts in Section 14 for quick estimation of both the simple and economizer cycles. These charts assume an isentropic efficiency of approximately 0.75 and so imply use of a centrifugal compressor.

The simple and economizer cycles solved above using hand calculations and a pure propane log P-h diagram are now simulated using OPSIM (App. 4). The refrigerant chosen for OPSIM simulation is similar to that of a refrigerant-grade propane—2.0 mol percent C2, 97.5 mol percent C3, and 0.5 mol percent iC4.

The following OPSIM computer results compare favorably with the previous hand calculations for both cycles.

Simple Cycle			**Economizer Cycle**		
Variable	Hand Calc.	Computer	Variable	Hand Calc.	Computer
m, Ib/hr	11363	11171	m_1, Ib/hr	2827	2950
-w, hp	268	260	m_2, Ib/hr	7874	7551
q_{cond},			hp_1	115	106
MMBtu/hr	1.68	1.66	hp_2	90	207
P_{cond}, psia	260	267.4	Total hp	205	207
P_{ch}, psia	32	31.2	q_{cond},		
			MMBtu/hr	1.52	1.52
			P_{econ}	117	117

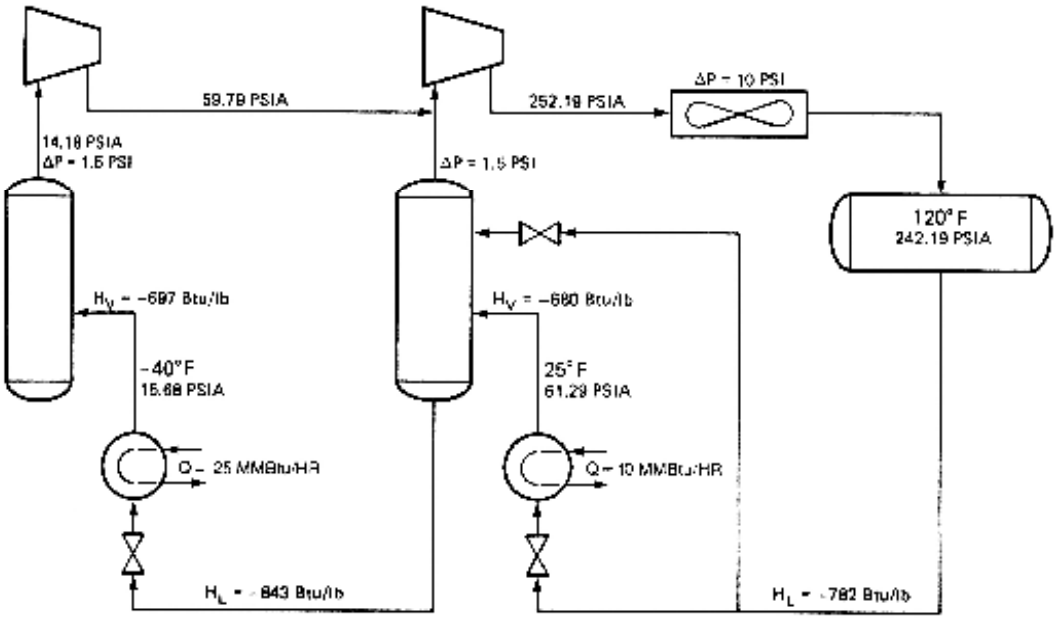


Figure 14-12. Two-stage refrigeration system (GPSA, 1987, Fig. 14-6).

Split-Level Refrigeration. A second potential improvement over the simple refrigeration cycle is encountered when two cooling duties are required at different temperature levels in a process, or when the duty can be divided into two parts, each done at a separate temperature level. Both duties can be provided by one two-stage refrigeration cycle as shown in Figure 14-12. The advantage of this arrangement is similar to that of the economizer cycle; only a portion of the refrigerant is circulated to the lower-pressure chiller, thus saving horsepower. On the other hand, two chillers are required rather than one, and the total area for heat exchange will be higher because the mean temperature driving force is less.

If split-level cooling is used on the front end of an expander plant, then the arrangement shown in Figure 14— 13 probably provides the best heat economy. Not only is a second chiller required but also a third gas-to-gas exchanger. Moreover, a two-stage unit requires compressor load-balancing control (Fraser, 1984).

Gas-engine-driven reciprocating compressors rather than centrifugal units power many, if not most, gas-processing refrigeration units. Also, two stages of compression are usually used because the high compression ratio would create too large a compressor rod loading. The example above has an overall compression ratio of 260/32 = 8.1, far above the usual limit of approximately 6.0. Thus, from a compressor standpoint, the addition of the economizer or use of split-level cooling involves very little additional investment—only one vessel or chiller and the previously mentioned load-balance control.

Fraser (1984) compared the split-level process and the economizer cycle with the simple cycle. As previously indicated, the economizer cycle is superior to the simple cycle. Eraser concludes that split-level refrigeration is a viable alternative in many plants, especially if the two refrigeration levels differ by more than 60°F and if cooling duties exceed about 3 MMBtu/hr.

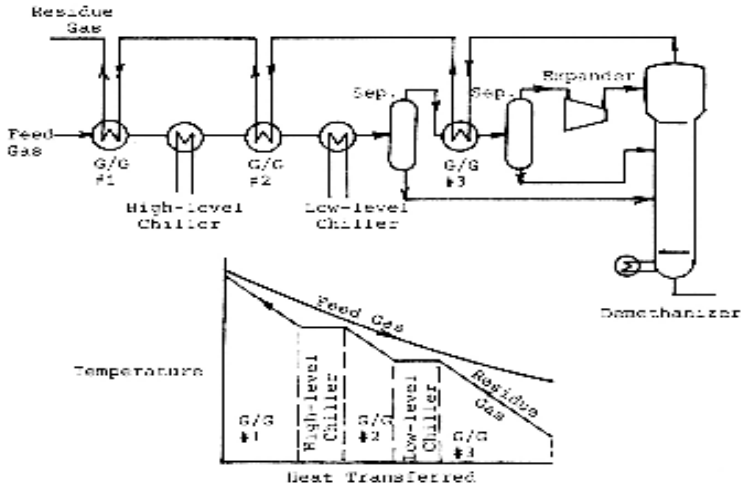


Figure 14-13. Split-level refrigeration for expander-plant feed: flow scheme and temperature pattern.

Apparently, split-level refrigeration is not commonly used. Blackburn and George (1979) show two such units. The process introduces complexity but has a large potential for saving energy costs.

Pure Refrigerant Limitations. Propane is not used for refrigeration below -40°F because its vapor pressure is less than one atmosphere. Air leakage into the system is a danger.

Cascade Refrigeration. As shown in Figure 14-14, refrigerant systems can be "stacked" to provide cooling below -40°F. The low stage uses methane as the working fluid. The methane is boiled in a chiller at near its atmospheric boiling point (-259°F), is compressed and condensed, rejecting heat to an ethylene evaporator at approximately -150°F. In turn, the ethylene is compressed and condensed, rejecting heat to a propane chiller at approximately -30°F. Finally, the propane rejects its heat to the surroundings. In this bootstrapping operation, heat can be absorbed at -250°F or so and rejected at ambient temperatures of about 100°F.

Cascade refrigeration has two drawbacks. First, a great deal of compression horsepower is required. Second, cascade systems are complex, with attendent operating and maintenance problems. A few two-stage systems are used in condensate recovery, but the process is not usually economically competitive. Cascade refrigeration is used in liquefied natural gas (LNG) plants.

Mixed Refrigerants. Geist (1985) describes the use of mixed refrigerants in LNG plants. Mixed refrigerant contains the hydrocarbons methane through pentane. This mixture evaporates over a wide range of temperatures rather than at one constant temperature. The vapor is compressed as one stream and rejects heat to cooling water or air. Compression requirements are thus lower than for a cascade system.

MacKenzie and Donnelly (1985) describe a mixed-refrigerant system used for condensate recovery. Further details are given later in the section on external refrigeration for condensate recovery.

Refrigerant Purity. As indicated in the previous OPSIM calculation, propane refrigerant used in gas plants is not pure, but typically contains small amounts of ethane and isobutane. Blackburn and George (1979) simulated three actual refrigeration cycles and concluded that ethane content of the refrigerant is particularly important. To control ethane content, they recommend purging, either at the cold end of the refrigerant condenser or on the vapor section of the surge tank.
The above simple refrigeration cycle is simulated using OPSIM for three different propane refrigerants having varying amounts of ethane and isobutane as impurities.

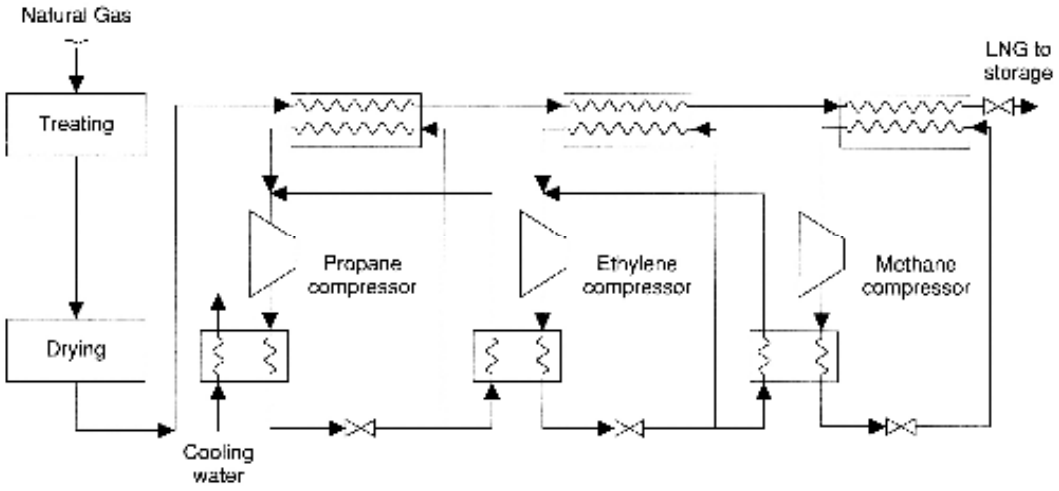


Figure 14-14. Cascade cycle (Geist, 1985).

The results tabulated below clearly confirm the undesirable effects of ethane and isobutane.

Refrigerant Composition, Mol %			Compressor
Ethane	Propane	Isobutane	Horsepower
2.0	97.5	0.5	260
4.0	95.5	0.5	266
2.0	96.5	1.5	262

Blackburn and George (1979) also note that compressor lubricating oil is a bothersome contaminant that can gather in the chiller and even coat the chiller tubes, thus reducing heat transfer. They recommend mounting the chiller on a slight angle from horizontal to facilitate drainage of the oil. The drained oil is sent to a reboiled separator to flash off propane and is then discarded.

External Refrigeration

Straight refrigeration for condensate recovery is a simple process. If propane refrigeration is used, the cold separator temperature is set straightforwardly.

If the hydrocarbon dew point is the controlling design parameter, then the feed gas must be refrigerated to a temperature low enough to provide a dew point lower than that encountered at any point in the downstream pipe line and/or the sales-gas specification. Practice is generally on the safe side. In other words, if a dew point of 30°F is desired, the gas will normally be cooled to a temperature safely lower than that required to obtained the desired hydrocarbon dew point. This allows for the possibility of retrograde condensation.

If maximum NGL recovery is desired, set the chiller temperature at the coldest practical with propane refrigeration, —20 to — 40°F, depending on the steel used. Russell (1977) noted that NGL recovery by refrigeration represents, in fact, a flexible process. As an example, he cites that the same plant can recover 60% of the entering ethane from a small flow of a rich gas stream (7 GPM C2+) or 30% of the ethane from three times as much of a leaner gas (3 GPM). Russell also shows how the effects of changing conditions can be estimated quickly.

Figure 14—15 shows the importance of the condensate stabilizer in basic refrigeration processes. The four variations shown differ in heat-exchange patterns and in disposal of the stabilizer overhead vapor. Types A and B differ only in the warming of the liquid feed to the top tray of the stabilizer. The warmer stabilizer overhead of type A requires liquid recycle to the feed, while the colder overhead of type B allows exchange with refrigerant and discharge to the residue gas.

The type C process features a refluxed stabilizer, which allows better product separation. Type D allows stabilization at a higher pressure (and thus temperature), and so removes the need for a recompressor.

In some gas plants the stabilizer overhead vapor is used as a fuel and therefore is not recycled or sent to the sales gas.

Russell states that for recovery of C3 +, types B and C are better than A at higher pressure and lower temperature, or for higher GPM. Type C is more appropriate for large plants, where its higher recovery efficiency is well utilized. For C2+ recovery, type B is favored over type A.

MacKenzie and Donnelly (1985) describe a mixed-refrigerant system for the recovery of 79% of the propane in a feed gas by refrigeration to — 100°F. For this particular case the mixed-refrigerant is less expensive to install and operate than an equivalent turboexpander plant. The plant is designed for a fairly lean feed gas (2.63 GPM), which yields less than 10% propane recovery when refrigerated to -30°F. In other words, ordinary propane refrigeration is not even a viable choice.

Figure 14—16 shows a mixed-refrigerant system. The inlet gas is refrigerated to -100°F and separated. The separator gas passes back through the gas-to-gas exchanger and thence to sales. The cold liquid is further heat-exchanged and fed to a deethanizer column. The bottom liquid goes to storage while the overhead vapor is recompressed, mixed with the cold separator gas, and sent to sales.

Mixed refrigerant is essentially 30/25/35/10 mol percent methane-ethane-propane-butane. The refrigerant flow is not described in detail, but Figure 14—16 reveals the basic pattern: partial condensation followed by expansion and vaporization.

Turbo-expander Plants

McKee (1977) describes the expander plant evolution from the early, simple plants to today's energy-efficient units. The expander plant shown in Figure 14-7 illustrates the operation very well but is somewhat misleading. Many variations occur in the heat recovery schemes (refrigeration recovery, actually). These and other items important to expander plant design and operation are now discussed, beginning with two important equipment items— turboexpanders and plate-fin heat exchangers.

Turboexpanders. Turboexpanders are rotary devices through which gas flows from high to low pressure. Work is recovered from the rotating shaft. An energy balance shows that this work is obtained from the decrease in gas enthalpy (if potential and kinetic energy changes are neglected and if the expander is considered adiabatic). This large decrease in enthalpy produces a correspondingly large temperature drop in the gas. This device produces low temperatures conducive to high condensate recovery.

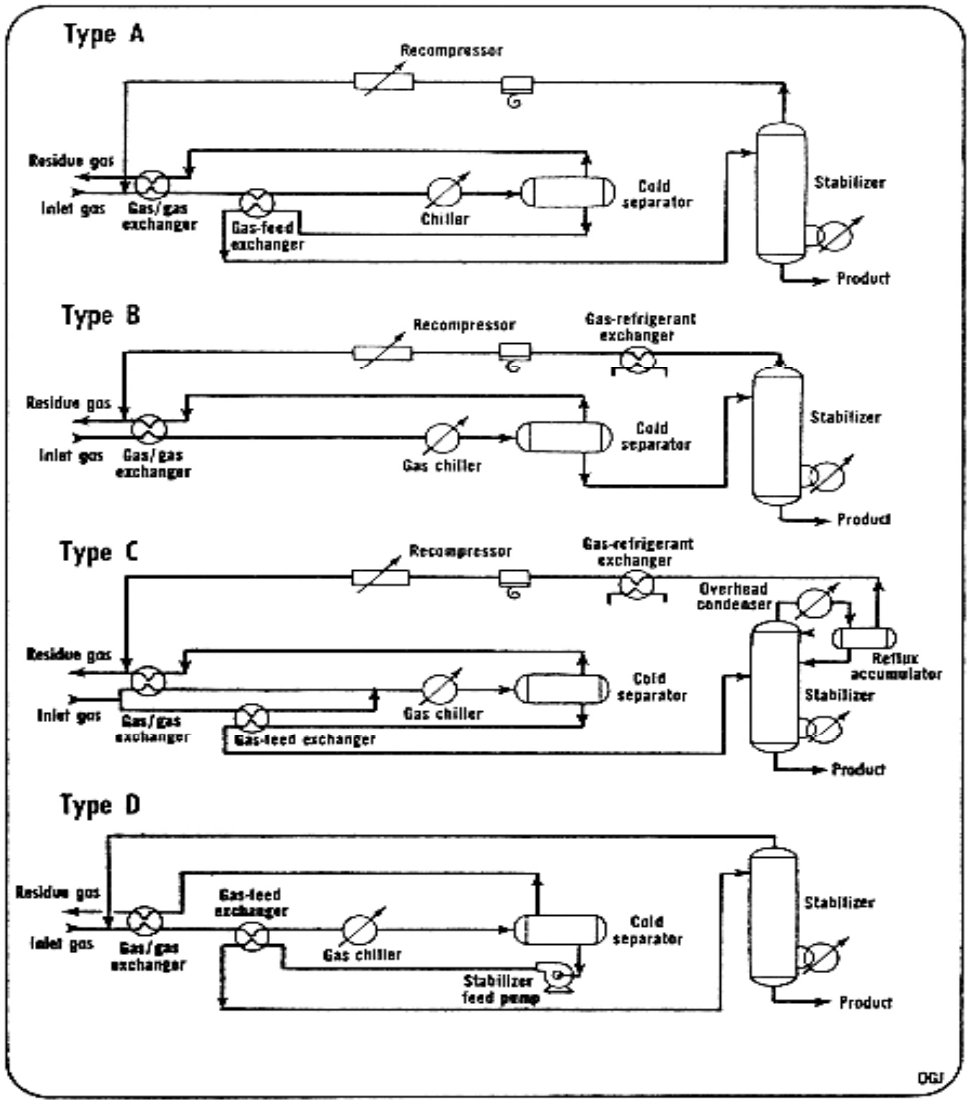


Figure 14-15. Refrigeration processes (Russell, 1977).

Figure 14-17 is a cross-sectional drawing of a typical turboexpander. Guided by adjustable inlet vanes, the feed gas flows radially inward in a tangential direction parallel to the rotor vanes. The gas spirals into the central area of the rotor (B) where another set of guide vanes directs the gas axially out without a rotating or vortex motion. Expanders are constructed to handle a condensing gas stream, with up to 10 weight percent liquid in the outlet (Valdes, 1984). However, solids such as solid CO2 erode the rotor and should be avoided.

In most expanders, the angle of the inlet guide vanes is adjustable. This allows the flow to be varied at a given inlet pressure, thus controlling the outlet pressure.

In Figure 14-18, as in most applications, the brake for the expander is a compressor, mounted on the same shaft. The unit is composed of a central bearing section with attached expander and compressor rotors. The expander and compressor covers can be removed easily for inspection and repair of the central section. The latter is the heart of the machine; it contains the impellers, seals, and bearings. This unit is easily replaced.

The expander shaft is stiff and rotates at 10 to 50 thousand or more RPM. The tip speed of the expander rotor is typically 1500 ft/sec (this and many of the other details listed below were discussed by McWilliams, 1985). The thrust and journal bearings are bronze and are oil-lubricated. The rotor is aluminum.

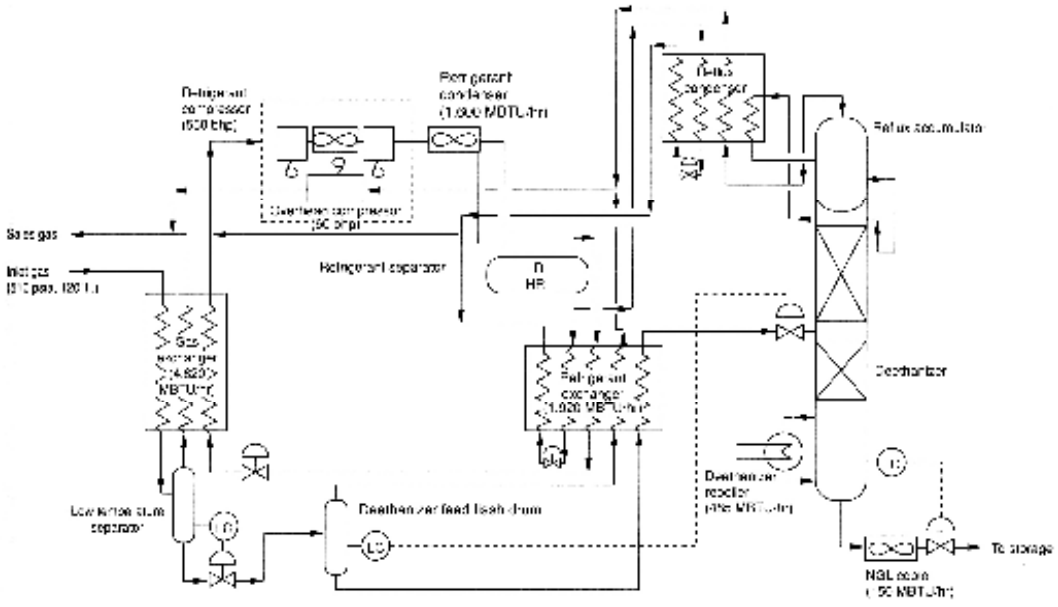


Figure 14-16. Mixed refrigerant flow (MacKenzie and Donnelly, 1985).

Seal gas is supplied to labyrinth seals at a pressure about 50 psi greater than the process stream, preventing oil leakage to the gas. At the same time the lubricating oil for the bearings is protected from the cold temperature of the process stream. The seal gas must be dry. If the seal gas supply fails, process gas goes into the oil system and could ruin the expander.

The start-up sequence for an expander is to (1) establish seal gas pressure, (2) start the oil lubricant flow, and (3) introduce the process gas. Of course shutdown is in the opposite sequence.

A channel between the two sides of the compressor impeller is used to equalize the pressure on the two sides. Axial thrust of the impeller shaft is thus minimized. Control of this thrust has been the biggest operating problem. Improvement of automatic control systems has gone a long way towards solving the problem (McWilliams, 1985).

Because quick shutdown is a must, the process-gas inlet valve must be capable of closing very quickly—on the order of 0.5 sec. Air-operated, spring-loaded valves are used.

Auxiliary systems needed are: a seal-gas supply system and a lube-oil recirculation system with filtering and cooling. The oil system can be equipped with an emergency feature that guarantees oil circulation for a short time while the expander is shut down.

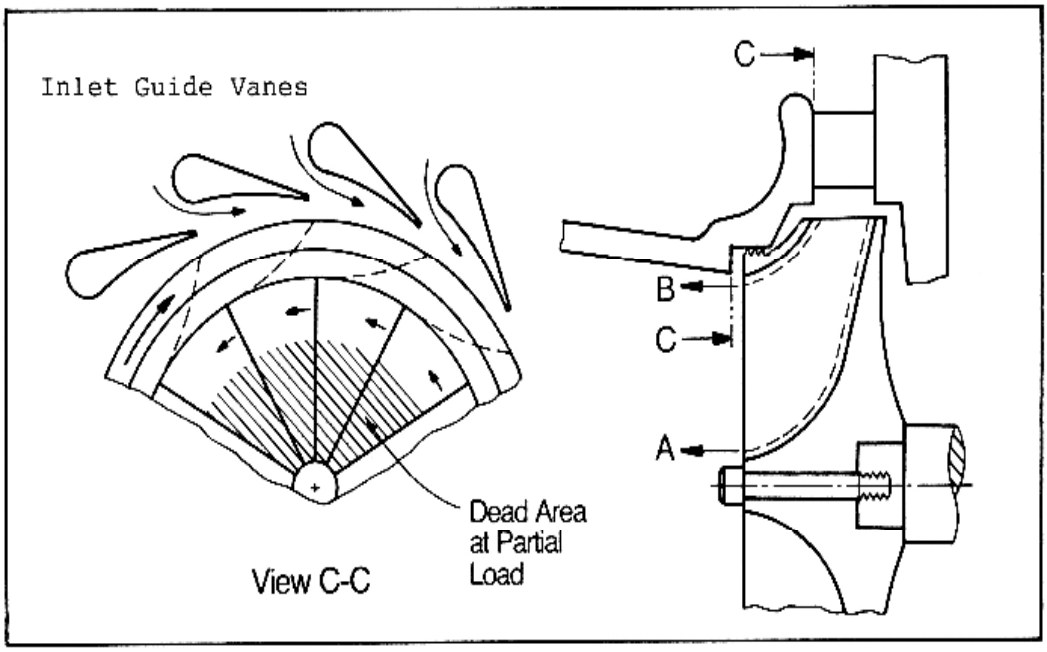
Operating problems include (GPSA, p. 13-42, 13-44):

- Ice or solid CO_2
- Abrasive solids
- Rotor resonance and bearing vibration
- Condensing streams
- Variable flow
- Oily feed gas
- High pressure
- Temperature shock

Special bearings handle resonance problems. This helps with variable flows. Channels in the rotor entry region help direct dust in the inlet stream to avoid problems from that source.

Expanders can handle services ranging from very low to as high as 10000 HP. One limitation is the expansion ratio per stage, normally 2.5 or less. However, expanders can be cascaded, with two or even three stages of expansion on the same or separate

shafts. Another limitation is the work per stage, which is usually confined to 50 Btu/lb or less.



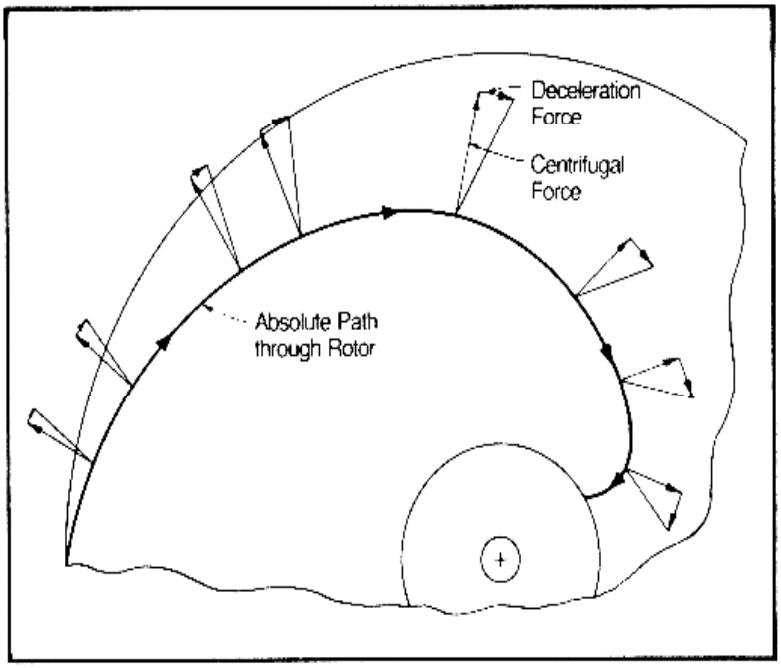


Figure 14-17. a) Cross section of turboexpander rotor and b) path of fluid in rotor

(Swearingen, 1972. Reproduced by permission of the American Institute of Chemical Engineers).

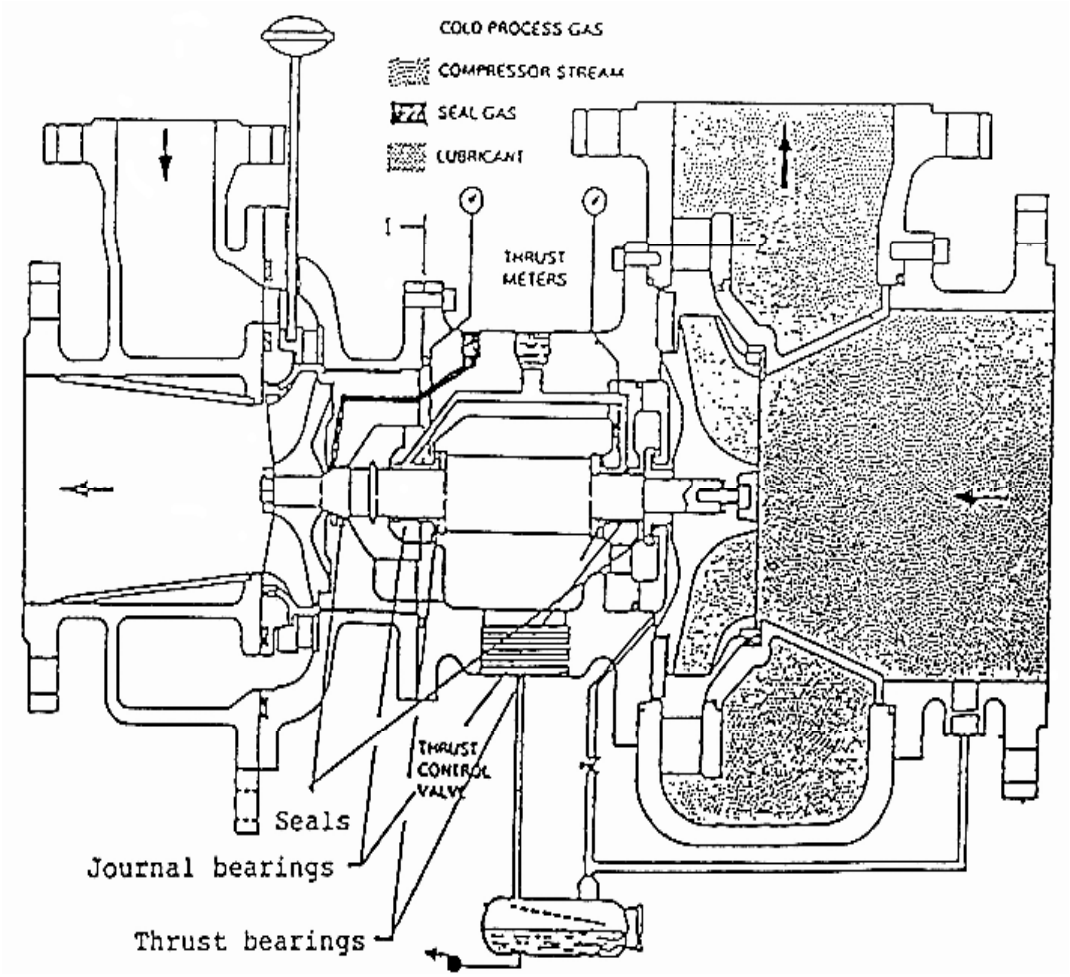


Figure 14-18. Schematic diagram of turboexpander (Holm, 1986. Reproduced by permission of the American Institute of Chemical Engineers).

As with compressors, expander design uses the isentropic model. This is easily illustrated by a pressure-enthalpy plot Figure 14-19. For expanders the reversible and adiabatic (isentropic) path slopes downwards and to the left on the p-H diagram. The enthalpy change for this path is the isentropic work. The actual work available from the gas is less than the ideal work because of irreversibilities in the gas as it flows through the expander.

The actual work recovered from the expander is thus the isentropic work times the isentropic efficiency. This efficiency is typically about 75%. As shown in Figure 14-19, the enthalpy change of the gas is less than the isentropic change. As a consequence, the temperature of the gas leaving the expander is somewhat higher than the isentropic value.

Not all of the work provided by the process fluid is available at the compressor end of the shaft because of friction in the bearings and seals. The process industry typically assumes that 98% of the work is recovered in the compressor.

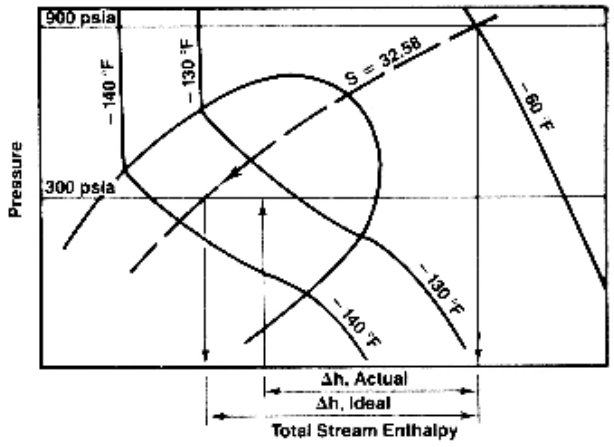


Figure 14-19. Turbo-expander model using pressure-enthalpy coordinates (GPSA, 1987, Fig. 13-63).

Plate-Fin Exchangers. An important type of heat exchanger used in expander plants is the plate-fin exchanger. These special-purpose exchangers (see Chapter 12, Fig. 12-14) are made of brazed aluminum, which can operate satisfactorily at very low temperatures. These exchangers are compact and efficient. Their chief drawback is the need to prevent solids from entering them. The narrow channels become clogged easily and the exchanger then performs poorly. Temperature shocks must be avoided also. Of course, shell-and-tube heat exchangers can be and are also used.

Design Procedure. The process objective is first assumed to be high percentage recovery of the entering ethane in the liquid product. Ethane rejection is discussed later.

Turboexpander or "cryogenic" plants operate with temperatures of -130 to -160°F at the top of the demethanizer tower. To avoid hydrate formation and damage to equipment, ***the feed gas must be dehydrated with solid desiccant***, usually molecular sieves. Also, expander plants must be thoroughly de-watered by evacuation or other procedure before commissioning and operation.

The following discussion is based on McKee's (1986) presentation. Other sources will be mentioned directly.
Start the design of a new plant by considering carefully the given information: pressure level, plant altitude, temperature level, flow rate, and composition.

The outlet pressure is ordinarily the delivery pressure to the sales-gas line, which ranges from 700 to 1000 psia, although occasionally it may be lower. If less than sales-gas delivery pressure, the inlet pressure is important because compression may be required to the expander inlet pressure.

Plant altitude is important because it has an appreciable effect on compressor driver horsepower, especially for gas turbine drives.

Feed-gas temperature is important since sometimes the feed can serve as the heat source to the demethanizer bottom reboiler. This saves energy and, at the same time, cools the inlet gas.

Flow rate is important for equipment sizing; the maximum expected rate should be used for this purpose. The lowest design rate also is important to the design of the recompressors. If the recompressors are centrifugal units, they have a definite surge limitation on the feed rate. If wide swings in feed rate must be provided for, it may be necessary to specify two parallel recompressors instead of one.

Composition is important for several reasons. First, CO_2 in the feed must be limited to, say, 0.5-1.0%. Otherwise solid CO_2 may form in the unit with subsequent equipment damage.

The condensate content is important also. High recovery of ethane (say 80%) without external refrigeration is possible only with a lean feed gas (up to 2.5 GPM) according to Wang (1985). Moderately rich gases require external refrigeration to obtain high recovery. And in the latter case, refrigeration horsepower is more effective than recompression horsepower (Wang, 1985). Rich gases will normally be processed by external refrigeration.

The next big decision is the percentage ethane recovery in the NGL. This may be dictated by external considerations, i.e., demand, profitability, management fiat. If the gas is very lean and contains inerts, then the sales-gas heating-value specification may control the ethane recovery. For lean gases at high cold-separator pressure, say

1000 psia, very little or no liquid may condense (in the upper retrograde region of the inlet-gas phase envelope), in which case the overall ethane recovery may be limited to 75-80%. One stage of turboexpansion will cool the feed gas enough to extract perhaps 80% of the ethane. Two expander stages will allow deeper extraction (say 90%) at the cost of increased recompression. Another technique for obtaining 90% recovery is the use of a refluxed demethanizer (see below).

A very important decision in expander-plant design is the recompressor. Compressors come in finite frame sizes, so that not just any pressure level can be chosen arbitrarily for the demethanizer. A compressor selection is made, and then the recompressor is calculated to determine its inlet pressure, from which the other important pressures can be selected (see McKee for details). McKee made definite recommendations for the isentropic efficiencies of the various rotating equipment:

Item	Isentropic Efficiency
Centrifugal recompressor:	
Small	0.72
Large	0.75
Reciprocating recompressor:	
High speed	0.80
Medium speed	0.82
Low speed	0.85
Refrigeration compressors	
Normal	0.70
Warmer	0.75
Expander	0.80

Refrigeration recovery is very important. If at all possible use the feed as the energy supply for reboiling. Feed temperature may preclude it as a heat source for bottom reboiling. On the other hand, side reboilers may do up to 60% of the total reboiling. These reboilers operate at very low temperatures on the process side as compared to ambient temperatures. Therefore it is possible to use feed gas as a heat source. This practice reduces the required refrigeration. Again, see McKee for details. Figure 14-20 shows a typical expander plant with side reboiling.

A simple example of the advantage of side reboiling is the sequencing of the inlet gas-to-gas exchangers in the absence of side reboiling. The two possible flow arrangements are shown in Figure 14-21. Note the position of the refrigeration chiller in the heat-exchanger train. Placing the chiller between the two gas-to-gas exchangers and the two side-reboiler exchangers allows the cold-separator temperature to be considerably lower than the chiller outlet temperature. This is a much more efficient use of the available refrigeration.

One expander plant variation presently used to a limited extent is the refluxed demethanizer (Fig. 14-22)—the flow sheet is proprietary and its use requires royalty payment. A portion of the chilled gas from the cold separator, say 30%, is bypassed around the expander and, instead, is cooled with demethanizer overhead vapor. The partly-condensed stream is then passed through a valve, further cooling and condensing it at a temperature lower than the outlet temperature of the expander. The bypass-stream, which is now colder than the expander outlet stream, goes into the top of the demethanizer as cold, mostly liquid reflux. The expander outlet stream enters at a lower point in the demethanizer. The net effect is higher C2+ recovery than is possible without the reflux, up to 90%.

Complete design of an expander plant involves trial and error. See McKee (1986) for details.

Ethane Rejection

Because the market price for ethane can vary widely, it is advantageous to have a versatile process capable of either retaining ethane or rejecting it. Such flexibility can be built into the expander plant as shown in Figure 14-23 (Valdes, 1984). In the ethane recovery mode, a portion of the inlet gas goes to the gas-to-gas exchanger, and the remainder through the product exchanger and the bottom and side reboilers. The fractionator operates as a demethanizer, its bottoms product being cold enough to provide refrigeration for the feed gas.

In the ethane rejection mode, all the inlet gas passes through the product exchanger and the bottom and side reboilers. External heating is provided for the feed gas in order that it can be warm enough to effect ethane rejection in the stabilizer. The gas then passes through the gas-togas exchanger, thence to the cold separator, etc. The source of heat for the feed gas can be the hot residue gas from the recompressor. Alternatively, hot oil can be used.

Evans (1980) outlines alternate ethane recovery/rejection schemes.

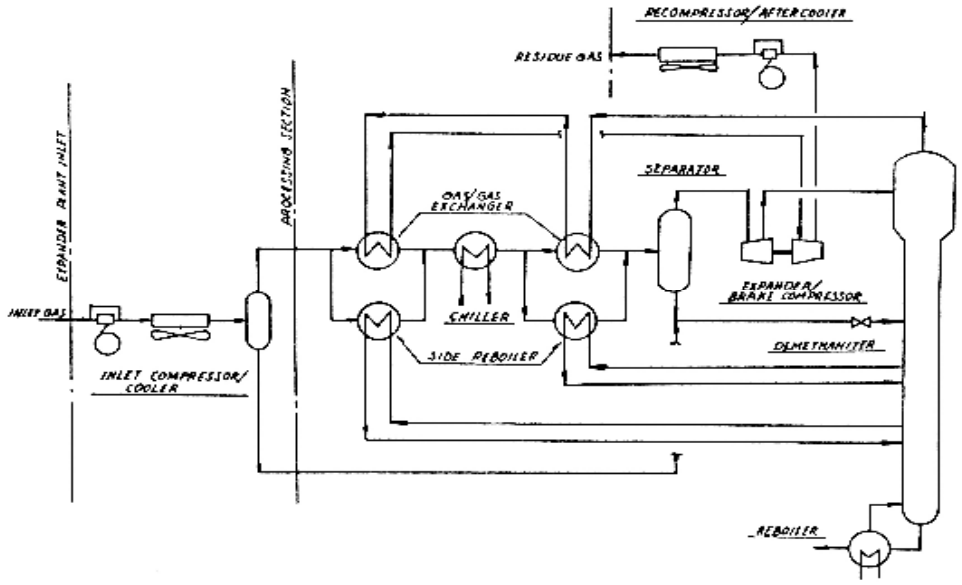


Figure 14-20. Expander plant with side reboiling (Wang, 1985).

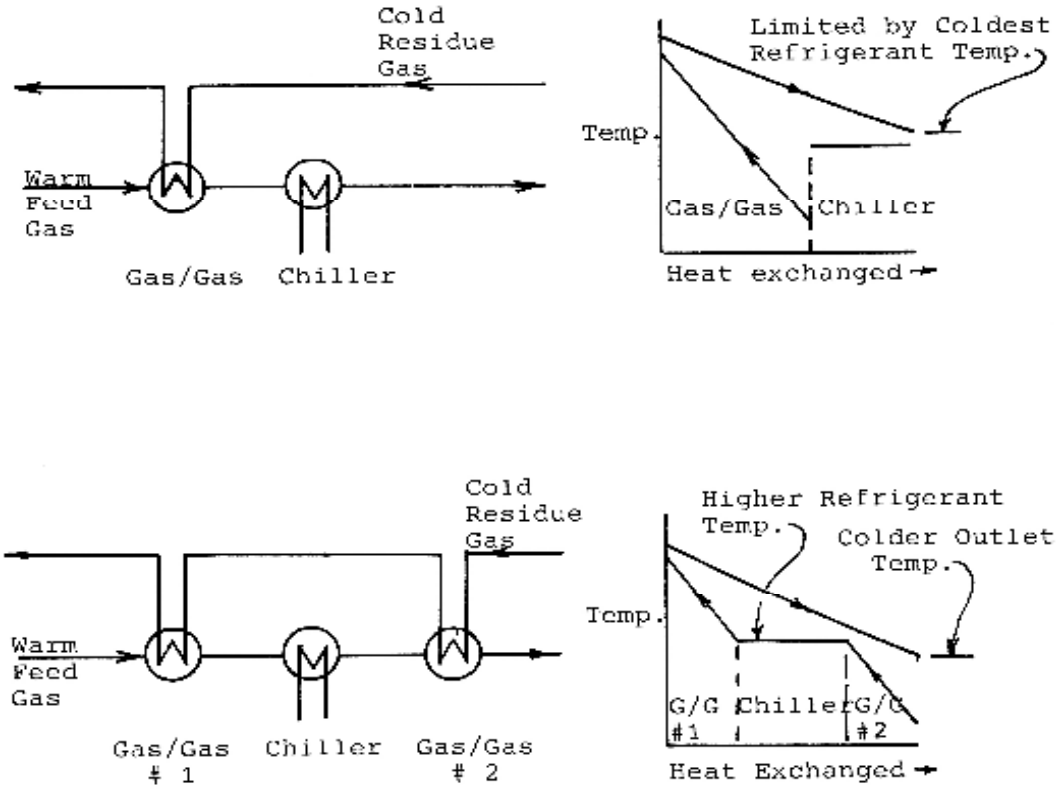


Figure 14-21. Gas-to-Gas/chiller arrangements.

Oil Absorption

Oil absorption plants are more complex than the simple depiction of Figure 14-8. An old plant, no longer in operation, but typical in its complexity is the Empress Plant in Alberta (Barber et al., 1965) shown in Figure 14-24. Some features of this plant are discussed to clarify the flow sheet.

First, note the rich-oil flash tank. This is fairly common in cold-oil plants. The idea is to heat the rich oil a bit and flash off some of the methane and ethane that would otherwise enter the rich-oil deethanizer (ROD), taking some of the load off that column. The flashed gas is recycled back to the plant feed gas.

Note next the series of three reboilers on the ROD (two are side reboilers). This arrangement is designed to maximize heat recovery from the hot lean oil, consistent with the temperature levels in the hot oil and the rich oil demethanizer.

The hot lean oil leaving the ROD reboilers passes through a series of two coolers (first air, then water). Then it passes through the rich-oil preheaters, being cooled to a temperature lower than ambient. These three coolers (air cooler, water cooler, and rich-oil preheater) minimize the load on the C3 chiller.

Note particularly that the lean oil next flows to the top of the ROD, where it contacts the methane-ethane-rich gas stream leaving the top of the ROD. This so-called lean-oil presaturator saturates the lean oil with methane and ethane before it is chilled and fed to the top of the absorber tower. This procedure avoids much of the temperature rise otherwise encountered on the top absorber tray due to absorption of methane and ethane. Presaturation also reduces the temperature in the whole tower, thereby allowing absorption of the desired propane with less lean oil flow than would otherwise be required. The lower the lean oil flow, the lower the heat duty of the still, and the lower the energy costs. Note: the still is the lean-oil fractionator.

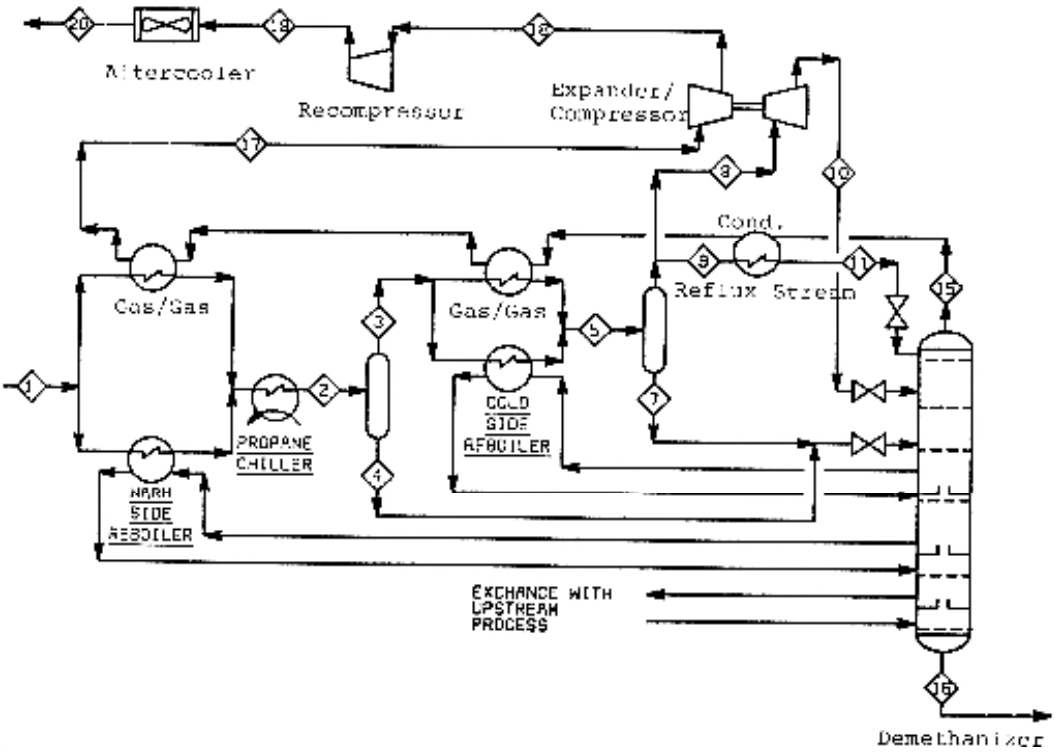


Figure 14-22. Expander plant with refluxed demethanizer.

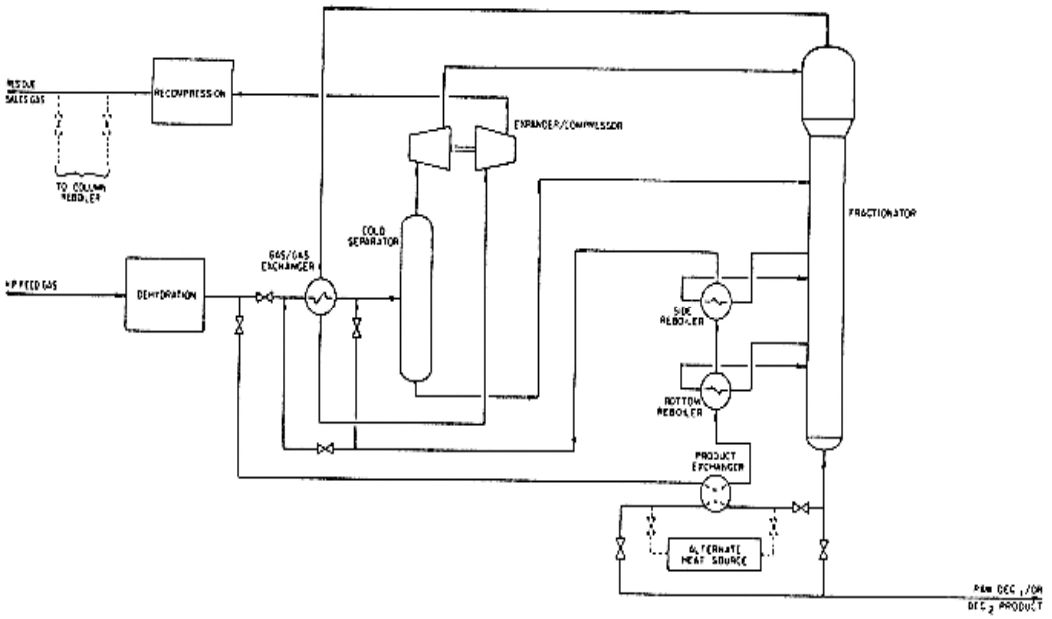


Figure 14-23. Turboexpander cycle designed for either ethane extraction or rejection (Valdes, 1984. Reprinted courtesy of Marcel Dekker, Inc.).

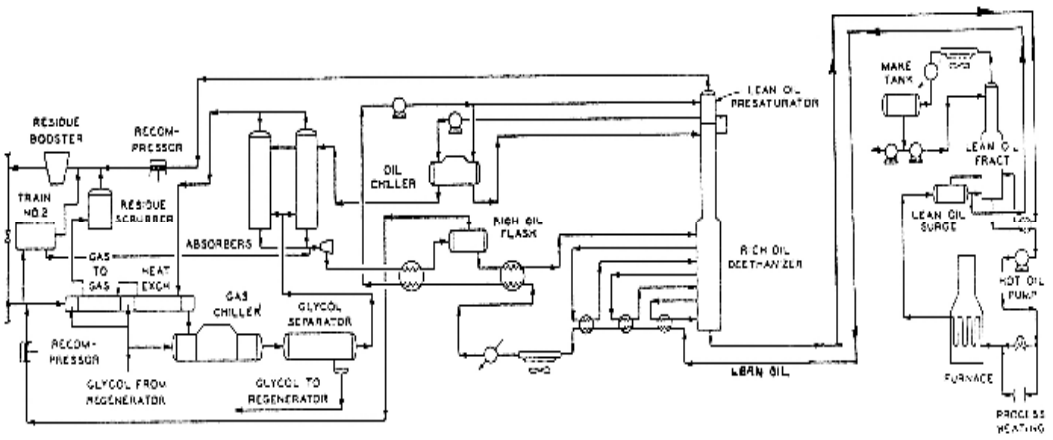


Figure 14-24. Empress cold-oil absorption plant (Barber et al., 1965).

The optimum pressure for the absorption towers depends on two factors: the inlet pressure of the plant feed gas and the phase behavior of the key component, propane. Observation of the effect of pressure on the vapor-liquid equilibrium K-value of propane (as on p. 25-93, GPSA, 1987) indicates that the optimum pressure will be in the range of 500-900 psia. The propane K-value at typical absorber temperatures goes through a minimum in this pressure range. If the pressure of the feed gas is low, it should be compressed to this range for absorption. If the feed pressure is at pipeline pressure, then the absorption should be done at that level to avoid recompressing the whole product gas stream.

The pressure of the ROD is determined by the separation of ethane and propane. If a ratio of 0.02 C_2/C_3 is used, then the composition of the ROD bottoms can be estimated closely. The temperature is set by consideration of the heat transfer to the lean oil from the still. A bubble-point calculation at the assigned temperature will set the pressure.

The lean-oil fractionator bottoms is taken to be at or near 550°F, a temperature which should not cause cracking of the lean oil in the furnace. A bubble-point calculation on the lean-oil composition at the assigned temperature yields the bottoms pressure.
In practice the true lean-oil composition is determined by the nature of the C6+ materials in the plant feed gas and how the still is operated. Regardless of the nature of the lean oil originally charged to the absorber plant after several days or weeks of steady state operation, the absorber oil will change to reflect the gas flowing into the plant and the still operation. The molecular weight of the lean oil should be kept as low as possible without producing excessive loss in the residue gas. A lean oil "reclaimer" is used (usually on a slip stream) to remove heavier molecular weight fractions from the lean oil. The lower the molecular weight, the lower the mass circulation rate for the same number of moles (which is what controls the absorption in the cold-oil absorber). Again, lower mass flow rate means better energy economy in the furnace.

Solid-Bed Adsorption

Barrere (1968) presents the following design equation for NGL recovery using solid-bed adsorption units.

$$w = 100\ n\ t\ y\ MW\ /\ L$$

where

w = the weight of adsorbent in the bed.
n = feed gas molar flow rate
t = cycle time, min (generally 2-4 hr)
y = mol fraction of key component in feed gas, usually propane
MW = molecular weight of key component
L = experimental loading of key component at breakthrough time, mass/mass

The bed volume is found by dividing the adsorbent weight by its apparent density. Judicious choice of bed length to diameter ratio provides uniform flow across the bed without excessive pressure drop. Barrere describes a method for determining the equilibrium capacity of the adsorbent material. See Chapter 9 for more details on solid desiccant adsorption.

Gas Processing in Enhanced Oil Recovery (EOR)

The recovery of crude oil from reservoirs can be enhanced in certain instances by injecting either nitrogen or carbon dioxide gas. Both of these processes have influenced gas processing. In both cases, the injected gas eventually shows up in the well stream. The produced gas therefore contains large amounts of nitrogen or carbon dioxide that lower the heating value below sales-gas specifications. Complex separation operations are required to reject N_2 and/or CO_2 from the hydrocarbon portion of the effluent gas.

Ryan-Holmes Process. A process suitable for CO_2 recovery is the Ryan-Holmes process (McCann et al., 1987). The difficulty with removing CO_2 from the hydrocarbon stream is the fact that the volatility of CO_2 lies intermediate to that of methane and ethane.

Figure 14—25 shows the process. Briefly, the low-pressure inlet gas containing hydrocarbon gas (including recoverable condensate) and $C0_2$ is dehydrated with glycol, compressed, and dehydrated thoroughly with mol sieves. The dry gas enters a chilled absorber where recycled C5+ oil absorbs propane and heavier hydrocarbons. The gas from the column is compressed and chilled to separate methane from CO_2 and ethane. The methane-rich stream has remaining CO_2 absorbed in the demethanizer, the overhead gas going to sales.

The rich absorber oil from the propane recovery column flows to, the additive recovery

unit, where it is separated into a light overhead stream and a bottoms stream suitable for recycling to the propane recovery column. This novel process separates the feed gas into residue gas, C_3 + condensate, and CO_2-C_2 streams. The latter stream is reinjected into the oil reservoir.

Nitrogen Rejection. Nitrogen rejection differs from the Ryan-Holmes process in that the condensate is first removed from the residue gas in conventional recovery processes. The residue gas now contains a large amount of nitrogen that would render the gas unsalable. Nitrogen rejection schemes allow separation of the hydrocarbon gases and nitrogen. The nitrogen may or may not be recycled to the oil reservoir.

Figure 14—26 shows a nitrogen rejection process (Alvarez et al., 1984). This cryogenic process involves extensive refrigeration economy. The feed gas is thoroughly dried with molecular sieves, then heat-exchanged with outgoing cold streams. Liquid is separated next and the gas is cooled further and introduced to the high-pressure column (approximately 400 psia). The high-pressure column is refluxed by condensing the overhead almost-pure nitrogen stream against vaporizing methane in the low-pressure column reboiler (approximately 20 psia). The crude methane stream from the bottom of the high-pressure tower is fed to the middle of the low-pressure column. The overhead nitrogen from the high-pressure column is heat-exchanged, then passes through an expander to produce a stream cold enough to provide reflux condensation for the low-pressure column. Almost pure hydrocarbon liquid is pumped from the bottom of the low-pressure column to sales-gas pressure, then exchanged with incoming streams to vaporize it. The total nitrogen from the low- and high-pressure columns is rejected to the atmosphere after heat exchange. Davis et al. (1989) describe in detail a very large combined NGL recovery/nitrogen rejection facility.

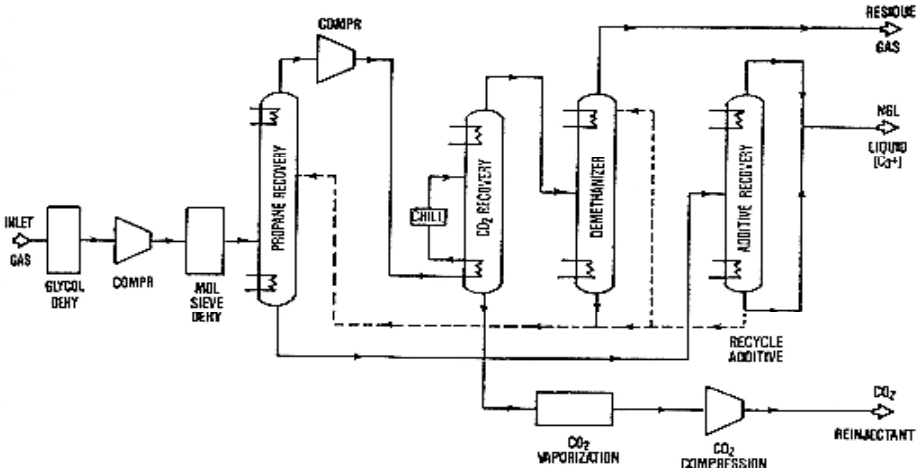


Figure 14-25. Ryan-Holmes process (McCann ef a/., 1987. Reproduced by permission of the American Institute of Chemical Engineers).

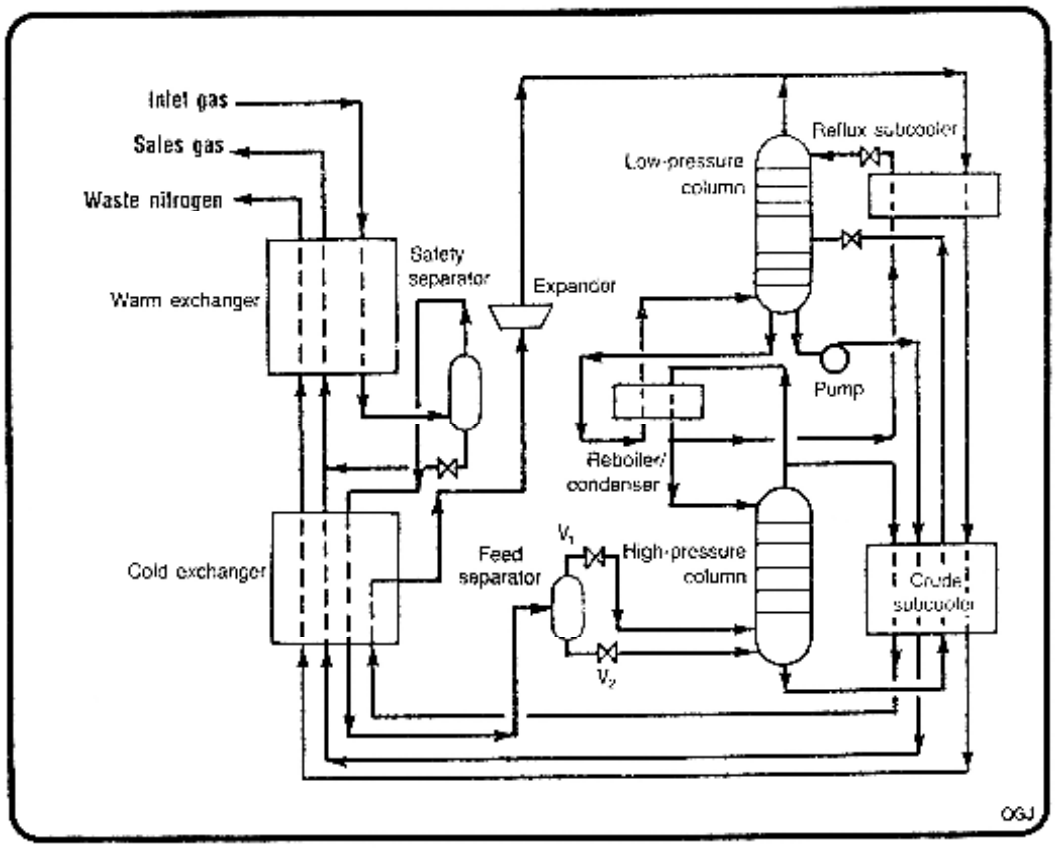


Figure 14-26. Two-column nitrogen rejection process (Alvarez ef al., 1984).

Helium Recovery. Because helium is very difficult to find and recover, it is one of the most expensive industrial gases. Processes used to recover helium from natural gases include: stand-alone cryogenic, recently developed standalone membrane units, and combinations of cryogenic, pressure swing adsorption (PSA), and membrane units (Choe et al., 1988)

LIQUID HANDLING AND TREATING

To this point, natural gas liquids have been referred to simply as a mixed liquid product. In older gas-processing plants, the usual practice was to have a fractionator train for separating the raw condensate into the usual products: propane, butanes (possibly separate i- and n-butane), and natural gasoline (C5 +). Present practice is generally to defer the fractionation of the raw NGL product to large central units. Raw product is pipelined or otherwise transported to the fractionation plant, where the individual products are produced. Storage at the production unit will be for a high-vapor-pressure product of broad boiling range. Treating for the removal of sulfur compounds is deferred also. The liquid is stored temporarily and pumped into the pipeline, truck, or tank car.

The separation of raw product into its constituents, the treating of these products, and their subsequent disposal is now addressed.

Fractionation

The raw liquid product is separated into individual products in a series of columns or towers in which the top product is the most volatile component in the feed. The first product to be removed, if present, is ethane. Columns for the removal of any component are referred to as "de-component-ers." The first column is the deethanizer or DEC2ER followed by the depropanizer or DEC3ER, and finally the debutanizer, DEC4ER. The bottoms from the debutanizer is the pentanes-plus (C5 +) or natural gasoline product. The butane tower overhead product commonly flows to a butane splitter, where the isobutane goes overhead and the normal butane to the bottoms. Each of the fractionated products is relatively pure. The actual purity depends on the sales specifications.

Sulfur compound removal and drying for water removal are required when sulfur-compound and water contents exceed the product specifications. Water is usually present, and the presence of sulfur compounds is determined by the feed gas analysis. If the feed gas is sour (contains H2S), mercaptans are likely to be present also.

Figure 14—27 shows the fractionation train used in a large, integrated plant that illustrates not only the essential elements but also a few added embellishments, such as the isobutane depropanizer and the propane deoiler. Note that the propane is treated and dried, as are the combined butanes. The natural gasoline is treated also. The fractionation sequence of Figure 14—27 will be the optimum one for most such plants. This particular plant has a deethanized feed; plants with ethane in the raw feed must have a deethanizer tower as well.

The distillation process takes advantage of the difference in volatility of the various hydrocarbons. Heat is introduced to the bottom of each column to drive the more volatile components present overhead into the distillate product. The towers are provided with condensed liquid reflux at the top to "knock back" the less volatile components into the bottoms product. Of course the distillate products are not pure but have some acceptable amount of the next-heavier component in them (heavier meaning less volatile).

Distillation columns contain trays with overflow weirs and downcomers. The reflux at the top of the tower flows across the top tray, into the downcomer, then across the next tray, etc. The vapor passes up through the tower through holes in the trays. In this manner, the vapor and liquid phases are thoroughly contacted, allowing the components to vaporize and condense easily and accomplish the desired separation.

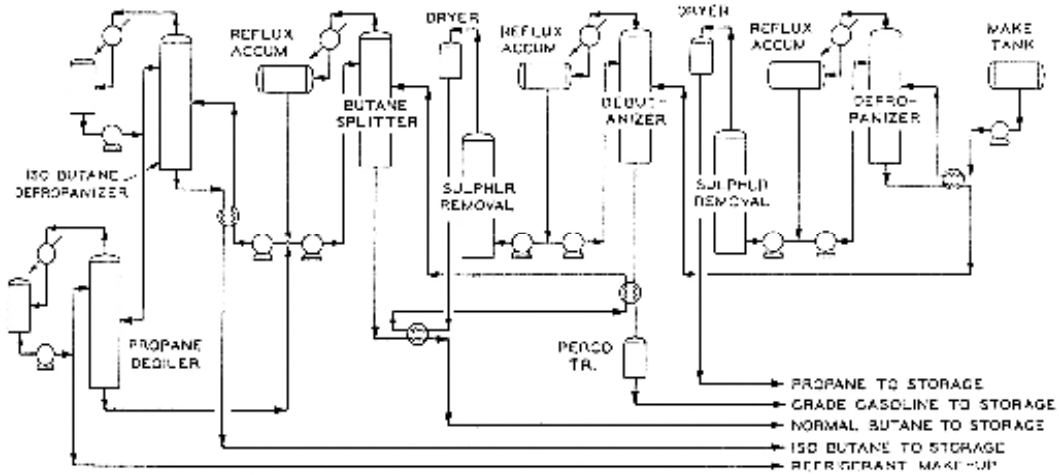


Figure 14-27. Product fractionator train (Barber ef a/., 1965).

Two types of trays are used commonly in fractionator plants: sieve trays and valve trays. Sieve trays are simply flat plates with holes punched in them. Valve trays have holes or slots into which movable plates are fastened. These plates, or valves, rise as vapor flows through them. Valve trays have wider operating ranges of vapor and liquid flow rate than do sieve trays.

Each column pressure is set by the temperature of the available cooling medium (cooling water or air). The tower pressure must be high enough to condense the reflux at a temperature a bit higher than that of the cooling medium (to allow a reasonable temperature approach in the condenser). A depropanizer operates at approximately 200 psia. Debutanizers and butane splitters operate at about 70 psia. If a deethanizer is used, the distillate product and/ or reflux will have to be condensed by use of propane refrigerant in order that the tower can operate at a pressure of about 400 psia. Separation in a distillation tower becomes more difficult as the

operating pressure goes up due to reduced relative volatility and density difference between the liquid and vapor.

Design of the distillation columns involves setting the number of trays and reflux rate, as well as the column diameter. The desired product purities determine the number of stages and reflux. The total feed rate controls the column diameter. The complex process calculations needed for this purpose are done routinely on the digital computer. Experience results in efficient design.

Treating

Tuttle and Allen (1976) approached treating liquid products by summarizing the products and their likely contaminants (Fig. 14-28). The volatilities or vapor-liquid equilibrium K-values of the various contaminants determine where they concentrate. Hydrogen sulfide, for example, has a volatility between ethane and propane and thus tends to appear in both the ethane and propane products. Predicting the likely contaminants in each product is a logical approach since the treating processes for the various sulfur compounds differ. Removal of hydrogen sulfide is done by a different method than removal of heavy mercaptans, etc.

Table 14-4 (Tuttle and Allen, 1976) summmaries the potential treating processes for the various contaminants. Masson and Davidson (1977), in their summary of treating methods used in Canada, state the aims of treatment in terms of the GPA specifications for propane and butane products:

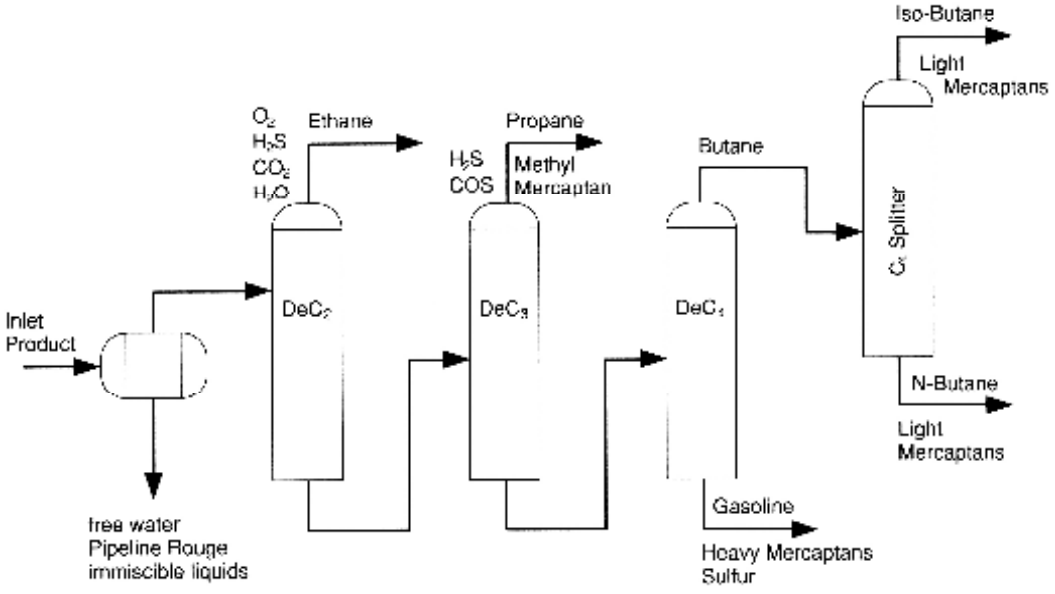


Figure 14-28. Fractionation plant contaminants (Tuttle and Allen, 1976).

Table 14-4 NGL Treating Processes (Tuttle and Allen, 1976)

				Containment			
				Mercaptans		Free	
	H_2O	CO_2	H_2S	Low MW	High MW	Sulfur	COS
Glycol:	X						
Counterflow							
Cocurrent	X						
Mole Sieve	X	X	X	X			X
Amine: MEA		X	X				X
DEA		X	X				X
Caustic: Wash		X	X	X		*X	
Regen'd.				X			
Merox: Extraction				X			
Sweetening				X	X		
Perco				X	X		

* If H_2S not present, free sulfur will not be removed and an Na_2S wash can be used.

- HD-5 propane specification:

Dryness—must meet the cobalt bromide test for water content
Volatile sulfur—10 grains per 100 cu ft, max Corrosion—No. 1 copper strip, max

- Butane specification:

Dryness—must contain no free water Volatile sulfur—15 grains per 100 cu ft, max Corrosion—No. 1 copper strip, max
Masson and Davidson (1977) summarize the processes used to remove the various contaminants and confirm the recommendations of Tuttle and Allen.

The processes referred to Table 14-4 are, for the most part, described in the GPS A Engineering Data Book (1987). Tuttle and Allen (1976) also furnish references for further study.

PUMPING AND STORAGE

Centrifugal pumps are generally used for liquid products. The usual care must be taken that product pumps be in-stalled with adequate suction head to avoid cavitation problems. Products are at their bubble-points in the surge or storage tanks; therefore any significant pressure drop in the flow lines below the bubble-point pressure will cause vaporization and pump cavitation.

Liquid-hydrocarbon storage is discussed in Section 8 of the GPS A Engineering Data Book (1987). Horizontal cylindrical vessels, either insulated or bare, are generally used. Insulated tanks are preferred for the more volatile products, such as a demethanized or deethanized raw product liquid.

Review Questions

1. Why should a NGL recovery plant be capable of either recovering or rejecting C2?
2. List the three objectives of NGL recovery processing.
3. What is the usual processing objective for a remote gas well? What are the expensive processing alternatives?
4. List the common NGL recovery processes.
5. Which NGL recovery processes are likely to be most attractive for:
   - lean gas
   - very rich gas
   - high pressure gas
   - low pressure gas
   - low production rates
   - remote locations?
6. How can a basic or simple refrigeration cycle be made more efficient?
7. In expander plants, cooling the feed gas with the residue gas is very important. Describe how this "refrigeration" can be made most efficient.
8. List some of the more common problems encountered while operating expanders.
9. What modifications are required for an expander plant to reject ethane?

Problems

1. Assume crude oil sells for $18 per bbl and natural gas for$1.50/Mscf.
   Estimate and compare the cost of energy in $/MMBtu for both crude oil and natural gas.
   List the major intangibles (no need to discuss).
2. Chapter 14 presents process calculations for the following refrigeration cycles: simple cycle, economizer cycle, and split-level refrigeration. These calculations assume that the propane refrigerant is condensed at 125°F (air cooling). Assume that suitable cooling water is available to condense the propane at 100°F.
   Estimate the efficiency of the simple, economizer, and split-level refrigeration cycles.

Nomenclature

C2+ = ethane and heavier hydrocarbons
C3+ = propane and heavier hydrocarbons
EPA = U.S. Environmental Protection Agency
GHV = gross heating value
h = enthalpy (Btu/lb)
HC = hydrocarbon
J-T = Joule-Thompson
L = loading of key component on solid bed, Ib/lb bed
LTS = low temperature separator
m = refrigerant circulation rate, Ib/hr
m_1 = vapor flow from economizer, Ib/hr
m_2 = liquid flow from economizer, Ib/hr
MTA = mass transfer agent, e.g., absorber oil or solid desiccant
MW = molecular weight, Ib/lb mol
n = feed gas flow rate, Ib mol/hr
P = pressure, psia
PSA = pressure swing adsorption
Q = specific heat duty, Btu/lb
q = heat duty, Btu/hr
W = compression work. Btu/lb
w = weight of adsorbent in bed, Ib
x = quality or mass fraction vapor
y = mol fraction of key component in feed gas

Subscripts

act = denotes actual work
ch = refers to chiller
cond = refers to condenser
econ = denotes most economical value
is = denotes isentropic (constant entropy) compression
m = denotes adiabatically mixed value
out = denotes outlet stream value
1, 2, 3, 4, 5, 6 = see Figures 14-10 and 14-11

MODULE 6:

Natural Gas Measurement

NATURAL GAS MEASUREMENT

In general, flow measurement is characterized by the continuous development of new and improved techniques. Therefore, the more common types of flowmeters, their current uses, and typical accuracies are reviewed in Volume 2.

Natural gas measurement is presently characterized by very extensive research programs worldwide that were, at least in part, undertaken because of differences between national and international standards. In the U.S., the American Petroleum Institute's Manual of Petroleum Measurement Standards (API, 1965-date) is the primary reference for all oilfield applications. Chapter 14, "Natural Gas Fluids Measurement," is of particular interest.

Orifice metering is described first in detail because orifice meters are almost always chosen for natural gas custody transfer both in the U.S. and internationally. Gas turbine meters have gained rapid and widespread use since being introduced in the U.S. in 1963. Accordingly, turbine meters are discussed next in detail. Vortex and ultrasonic meters find limited but increasing use, so they are reviewed also. Critical flow nozzles or sonic nozzles are often used to calibrate or prove meters and are described in Volume 2. Some selection criteria—relative advantages, disadvantages, and uncertainties of measurement—are then presented for orifice, turbine, vortex, and ultrasonic meters. Because natural gas is often odorized when metered and transported, Chapter 11 concludes with a brief summary of odorization.

ORIFICE METERS

An orifice plate is essentially a restriction in the pipe that forces the fluid to accelerate and then decelerate as it flows through the meter. Figure 11-1 shows the resulting changes in pressure as confirmed by the Bernoulli equation (ANSI/ API, 1985). Flow rate is inferred from the pressure difference measured by pressure taps upstream and downstream of the plate. Figure 11-1 also shows that the observed pressure drop depends on the tap locations. While Table 11-1 shows the locations of the more commonly used taps, flange taps dominate in the U.S., and flange and radius (or D and D/2) are common in Europe.

Orifice metering is now described in detail using the following topics: meter piping and differential pressure (DP) devices, meter tube and fittings, sealing units, ANSI/API specifications, calculation of gas flowrates, orifice meter operation, sources of error, and current research programs.

Meter Piping and DP Devices

Figure 11-2 shows a typical arrangement for gas meter piping. Valves A and B are bypass valves, C the bleed valve, and D and E supply valves. While Figure 11-2 may suggest that the meter is "clamped" to the pipe, a separate, firm, reasonably level, and vibration-free support is highly recommended.

Figure 11-3 shows schematics of the mercury-manometer and bellows-type differential-pressure (DP) measuring devices. The bellows-type is often preferred because:

1. It is particularly adaptable to measuring wet gas due to its self-draining feature.
2. It finds wide applicability with integrators and controllers due to its rapid response and high torque output.

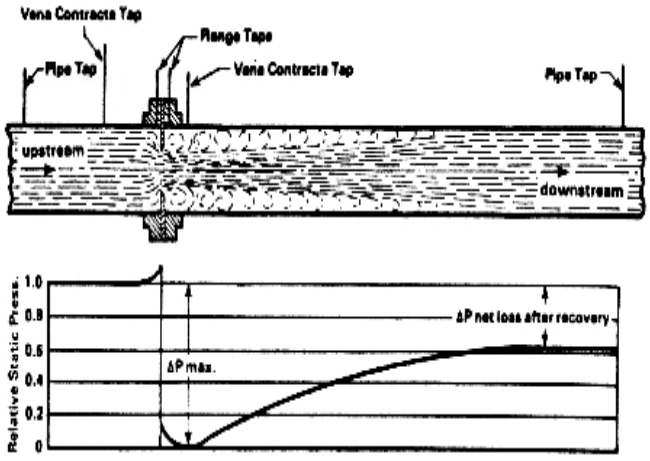


Figure 11-1

Flow and pressure patterns (North American Mfg. Co., 1978).

Table 11-1 Pressure Taps (North American Mfg Co., 1978)

Type	Upstream tap location	Downstream tap location	Advantages and disadvantages
Flange taps	1" from upstream face	1" from outlet face	Convenient tap locations fixed by purchasable flanges. Difficult if such flanges are not available
Vena contracta taps	1 x D from upstream face	d/D□ / from downstream face: 0.3 — 0.80 x D; 0.4 — 0.74 x D; 0.5 — 0.66 x D; 0.6 — 0.57 x D; 0.7 — 0.45 x D	Most sensitive (highest pressure differential). Fussy tap locations.
Radius taps	1 x D from upstream face	½pipe diameter from upstream face	Easier to locate than vena contracta. Meager data available on flow coefficients.
Pipe taps	2½x D from upstream face	8 pipe diameters from upstream face	Not dependent on d/D ratio. Pressure drop = net pressure loss. Take up a lot of space.
Corner	upstream edge	downstream edge	

The bellows-type meter should always be mounted above the orifice to exploit its self-draining ability, unless a liquid-filled system with seal pots, etc., is used.

Meter Tube and Fittings

Fig. 11-4 shows typical meter tubes for orifices.
A senior orifice fitting (Figure 1 l-5a) consists essentially of two chambers separated by a lubricated slide valve. This permits removal of the orifice plate without flow interruption or bypass around the meter. Senior fittings are commonly used for custody transfer or sales meters where removal, inspection, and changing of orifice plates are required and where flow bypass is illegal.

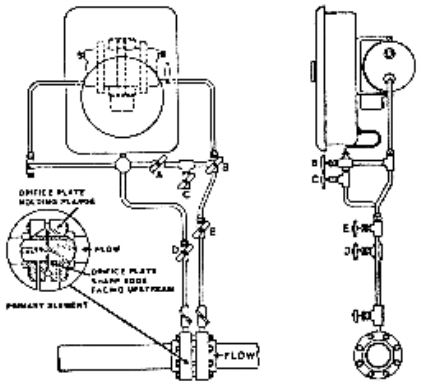


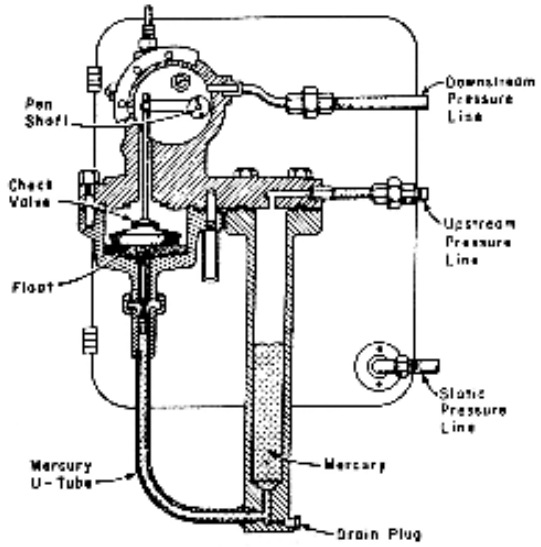
Figure 11-2
Meter piping for gas (Crabtree, 1981).

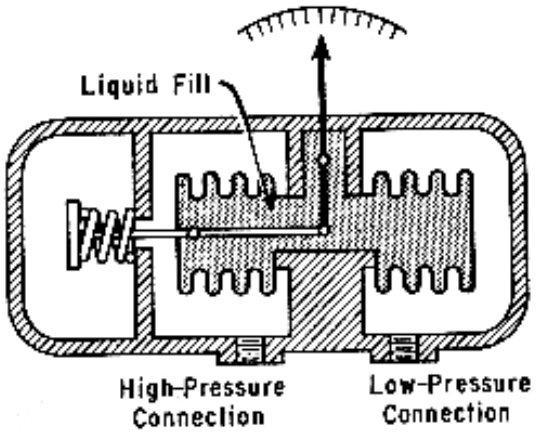
The junior orifice fitting (Fig. 11-5b) uses the same type of plate carrier as the senior fitting, and the orifice plate is removed similarly by a rack and pinion. Because there is no upper chamber, the fitting must be depressurized using block and blowdown valves before removing the orifice plate.

The simplex orifice fitting is a single chamber device (Fig. ll-5c). Removing the orifice plate requires depressurization of the fitting by loosening the top set screws, sliding the clamping bar out, and lifting the sealing bar carrier and orifice plate.
Standard orifice flanges are still widely used to hold orifice plates; however, plate removal and replacement are time-consuming.

Sealing Units (Fig. 11-6)

If a plate seal is used, then the dual sealing (DS) unit is the most common. This seal consists of a molded Hycar synthetic rubber ring. A slot in the inner center holds the round universal plate firmly. In larger sizes (12 in. +) the seal is vulcanized to each side of the plate and is called a dual vulcanized seal (DVS).





Mercury-Manometer-Type Differential-pressure Measuring Device	Schematic View of Bellows- Type Differential-pressure Measuring Device

Figure 11-3
Differential pressure devices (PETEX, 1987. Copyright © 1987 by the University of Texas at Austin).

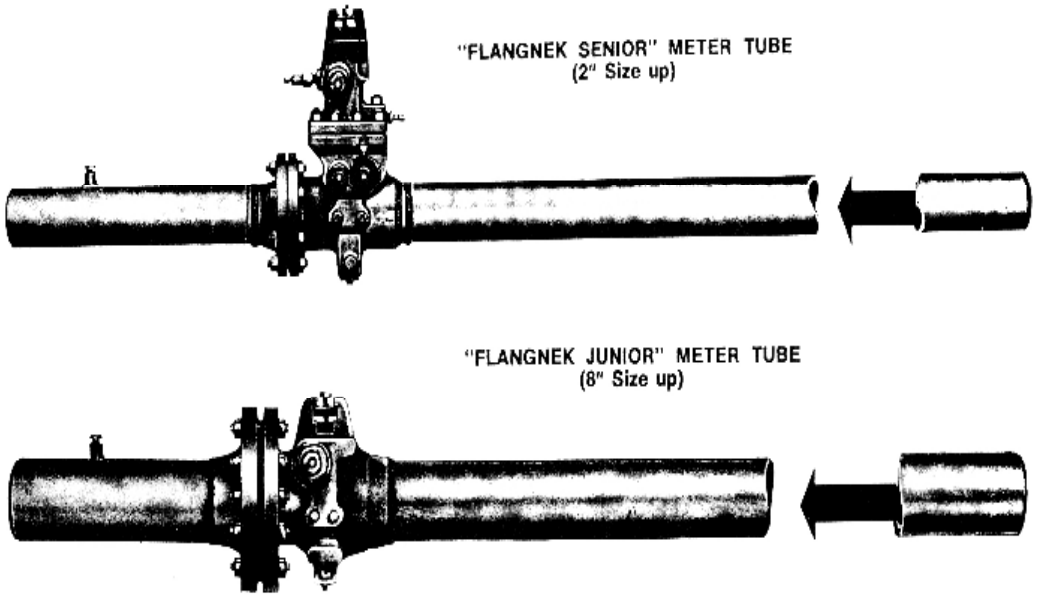


Figure 11-4
Typical meter tubes (Daniel, 1983).

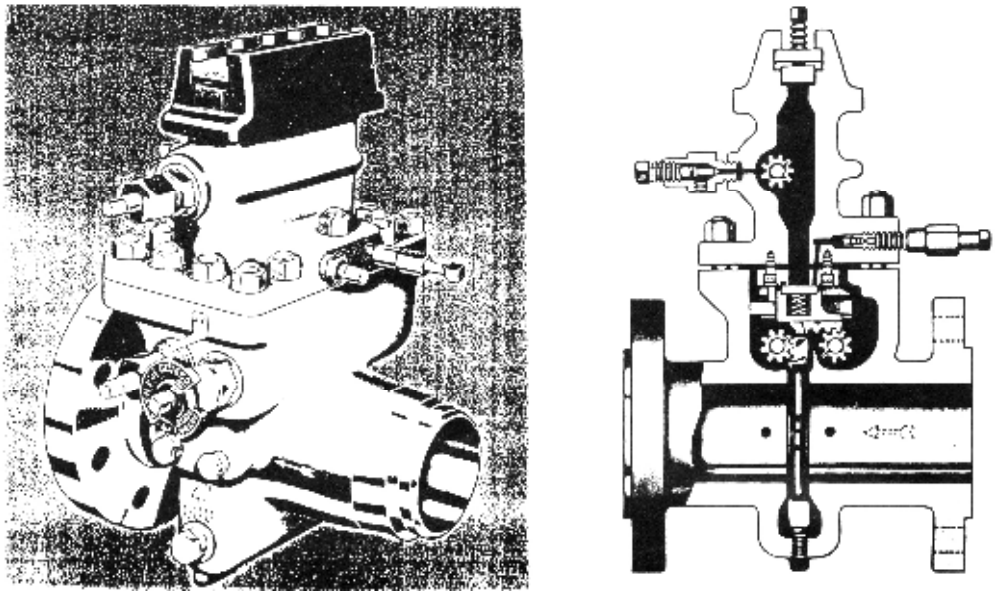


Figure 11-5a
Senior orifice fitting (Daniel, 1983).

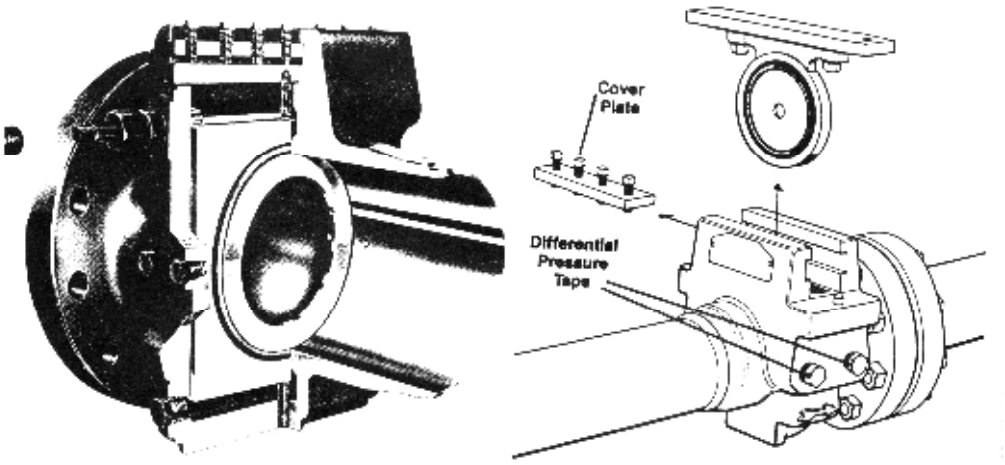


Figure11 5b.Junior fitting (Daniel, 1983).	Figure11 5c.Simplex fitting (GPSA, 1987).

Figure 11-6
Orifice seals (Daniel, 1983).

At higher temperatures (275°F) a metal (MS) or a Teflon (TS) seal should be used. These seals are two-piece seals that are attached to the orifice plate by spring clips.
The snap seal ring consists of two metal rings having an O-ring on both sides. These two metal rings are held onto the orifice plate by a specially designed external snap ring.
ANSI/API Orifice Specifications

Because orifice plates are NOT CALIBRATED they must be manufactured and installed to meet very definite specifications. In the U.S., the standard is ANSI/API Standard 2530 (1985). It also is referred to as AGA Report No. 3, GPA 8185-85, or API Manual of Petroleum Measurement (Chapter 14, Section 3).

Orifice Plate. The plate is usually made of 304 or 316 stainless steel or monel according to the specifications listed in Figure 11-7. The upstream edge shall be flat to within 0.01 in. per inch of the dam height—(D - d)/2. The upstream edge shall be square and sharp so that it does not show any light when checked with an orifice edge gage (Bean, 1985), or it does not reflect a beam of light. Unofficially, a fingernail may be drawn across the orifice edge—a sharp edge will catch or "bite."

The orifice must be concentric within 3% of the metering tube inside diameter.

The mean orifice diameter, d, is defined as the arithmetic average of four or more evenly spaced inside diameter measurements. Practical tolerances are:

Orifice Diameter	Tolerance, plus or minus
0.250 in.	0.0003 in.
0.375	0.0004
0.500	0.0005
0.625	0.0005
0.750	0.0005
0.875	0.0005
1.000	0.0005
over 1.000	0.0005 per in. of d

The Beta Ratio, or orifice (d) to meter (D) diameter ratio shall be flange taps $0.15 < ß < 0.70$ pipe taps $0.20 < ß < 0.67$

Meter Tube. The inside pipe walls shall be smooth (less than 300 microinches roughness). Grooves, scoring, pits, ridges, and offsets are not allowed.
The allowable tolerances of the actual meter tube ID to the published ID varies from 0.7% at ß = 0.7 to 4.4% at ß = 0.15 for flange taps (e.g., GPSA, 1987, p. 3-8).

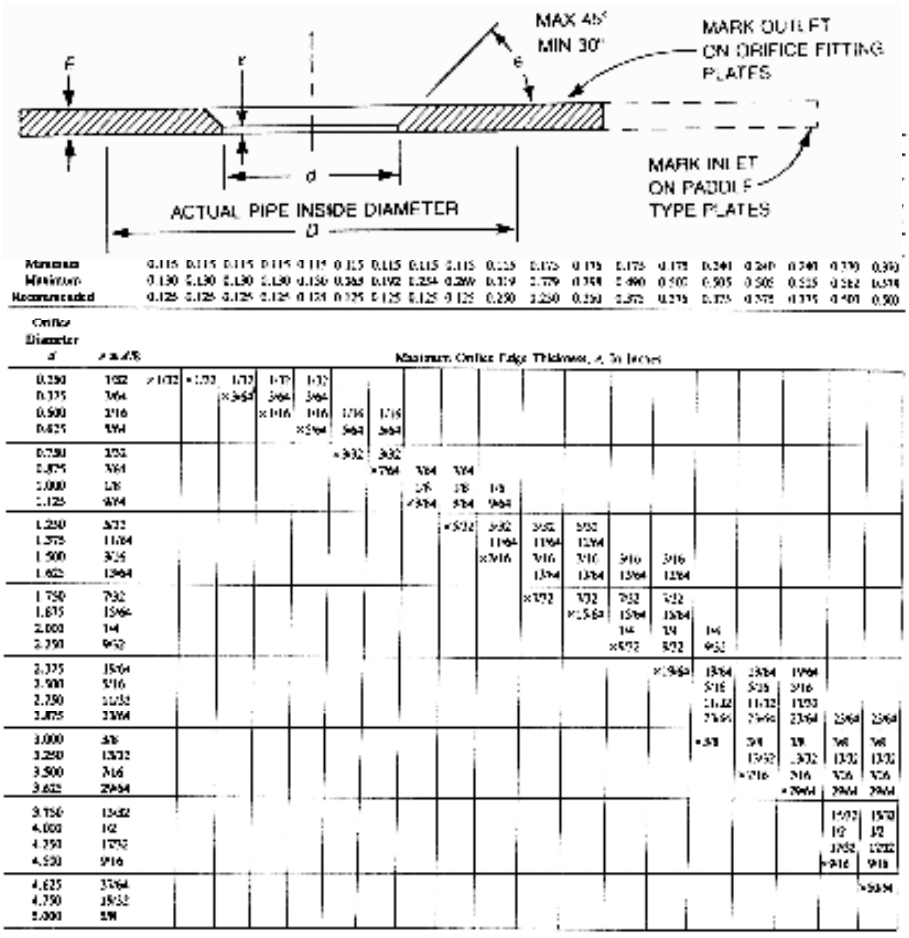


Figure 11–7. Orifice plate dimensions (ANSI/API, 1985).

Figure 11-7
Orifice plate dimensions (ANSI/API, 1985).

Length of pipe preceding and following orifice varies with the type of fittings (partially closed valve, ell, reducer, or expander) in the line—see ANSI/API, or GPSA (1987, p. 3-8, 3-9) for numerical values. For safety assume that (ß = 0.7 when designing new installations. Straightening vanes (e.g., GPSA, 1987, p. 3-11) reduce the required lengths of upstream pipe, and this is especially welcome for large diameter pipe when space is limited, e.g., offshore. Occasionally, vanes can cause plugging and slipping problems.
Other Specifications. Refer to ANSI/API 2530 for additional details on straightening vanes, orifice flanges, orifice fittings, pressure tap holes, and thermometer wells.

Calculating Gas Flow Rate

As defined in ANSI/API 2530 (1985), the fundamental equation for gas measurement is:

$$Q_h = C' \sqrt{h_w p_f} \qquad (11\text{-}1)$$

where

Q_h = quantity rate of flow at base conditions in cu ft/hr
h_w = differential pressure in inches of water
P_f = absolute static pressure, psia
C' = orifice flow constant

$$C' = F_b.F_r.Y.F_{pb}.F_{tb}.F_{tf}.F_{gr}.F_{pv}.F_a.F_{am}.F_{w1}.F_{wt}.F_{pw1}.F_{hgm}.F_{hgt} \qquad (11\text{-}2)$$

And

F_b = basic orifice factor
F_r = Reynolds number factor
Y = expansion factor
F_{pb} = pressure base factor
F_{tb} = temperature base factor
F_{tf} = flowing temperature factor
F_{gr} = specific gravity factor
F_{pv} = supercompressibility factor
F_a = orifice thermal expansion factor
F_{am} = correction for air over the water in the water manometer during the differential instrument calibration
F_{wl} = local gravitational correction for water column calibration standard
F_{wt} = water weight correction (temperature) for water column calibration standard
F_{pw1} = local gravitational correction for dead weight tester static pressure standard
F_{hgm} = manometer factor, correction for gas column in mercury manometers
F_{hgt} = mercury manometer temperature factor

Brief descriptions of these factors follow, for more complete equations, tables, and descriptions see ANSI/API 2530.

Basic Orifice Flow Factor, F_b, is the scf of air that will flow per hour through the orifice at 60°F with a static pressure of 14.73 psia and a differential pressure of 1 in. water column. Values of Fh, which depend only on d, D, and the choice of flange taps, are given in ANSI/API 2530 and GPSA (1987, p. 3-14, 3-15).

Reynolds Number Factor, F_r, corrects for changes in C' with flowrate or Reynolds number. Values for F_r ($F_r = 1 + b/\sqrt{h_w p_f}$) are given in GPSA (1987, p. 3-16, 3-17).

Expansion Factor, Y, adjusts C' depending on whether P(is measured from the upstream or downstream pressure tap. See GPSA (1987, p. 3-18, 3-19) for values of Y.

Pressure Base Factor, F_{pb} adjusts the flow (scfh) from the AGA base of 14.73 psia to any other base pressure (Ph). Numerically,

$$F_{pb} = 14.73 \; / \; P_b \text{ (psia)} \qquad (11\text{-}3)$$

Temperature Base Factor, F_{tb}, adjusts the flow (scfh) from the AGA base of 60°F to any other temperature base, T_b (°F). Numerically,

$$F_{tb} = (460 + T_b) \; / \; (520) \qquad (11\text{-}4)$$

Flowing Temperature Factor, F_{tf}, adjusts Q if the flowing temperature, T_f (°F), is not 60° F. Numerically,

$$F_{tf} = \sqrt{(520)/(460 + T_f)} \qquad (11\text{-}5)$$

Specific Gravity Factor, F_{gr}, corrects the flow rate when the specific gravity of the flowing gas, SG, is not that of air. Numerically,

$$F_{gr} = \sqrt{1/SG} \qquad (11\text{-}6)$$

Supercompressibility Factor, F_{pv}. ANSI/API describes a detailed, empirical procedure for calculating F_{pv} which corrects for real gas behavior (compressibility factor, Z≠1). Numerically,

$$F_{pv} = \sqrt{1/Z} \qquad (11\text{-}7)$$

Orifice Thermal Expansion Factor, F_a, corrects for thermal expansion of the orifice plate. For 304 and 316 stainless steel:

$$F_a = 1 + 0.0000185 \; (T_f - T_{meas}) \qquad (11\text{-}8)$$

For monel:

$$F_a = 1 + 0.0000159 \; (T_f - T_{meas}) \qquad (11\text{-}9)$$

where

T_f = flowing temperature (°F)

T_{meas} = temperature (°F) at which orifice diameter is measured.

Water Manometer Correction Factor, F_{am}, corrects for the gas leg over water when the differential gage is calibrated using a water manometer

$$F_{am} = \{[62.3663 - (P_{atm} + h_w/27.707)/192.4]/62.3663\}^{0.5} \quad (1\ 1\text{-}10)$$

F_{am} = 0.99924 when P_{atm} = 14.73 psia and h_w = 100 in. Local Gravitational Correction for U-Tube Manometers, FWJ. The local weight of a manometer fluid is directly proportional to the local gravity force.

$$F_{wl} = (g/32.17405)^{0.5} \quad (11\text{-}11)$$

In turn, g may be estimated by (ANSI/API 2530):

$$g = 0.032808*(978.018\ 55 - 0.0028\ 247\ L + 0.002\ 029\ 9\ L_2 - 0.000\ 015\ 058\ L_3 - 0.000\ 094\ H) \quad (11\text{-}12)$$

where

L = latitude in degrees

H = elevation in feet above sea level.

Water Manometer Temperature Correction Factor, F_{wt}, corrects for variations in the density of the water in the manometer used to calibrate the differential gage

$$F_{wt} = (\rho w / 62.3663)^{0.5}$$

where p_w = density of water (lb/ft^3) in manometer.

Local Gravitational Correction for Weight Standards, F_{pwl}, corrects for the local gravity effect on the weights of the dead weight tester

$$F_{pw1} = (g/g_o)^{0.5} \quad (11.13)$$

Where

g = acceleration due to local gravity force

g_o = acceleration of gravity used to calibrate dead weight tester.

The Mercury Manometer Factors, F_{hgm} and F_{hgt}, correct for the gas or vapor leg and the temperature respectively. See ANSI/API 2530 for definitions. Omit F_{hgm} and F_{hgt} for bellows meters.

Example 11-1. Calculate meter coefficient and gas flowrate. An orifice meter run has the following characteristics:

Meter Pipe 4 in. schedule 40 (4.026 in. ID)flange taps
static pressure measured from upstream tap

Orifice Plate stainless steel
1.500 inch at 20°C(68°F)

Recorder 100 in. WC differential
1000 psia static spring

Readings atmospheric pressure = 14.4 psia (500 ft elevation)

flowing temperature = 100°F
gas gravity = 0.60
static pressure = 651 psig
differential = in (bellows)

Base Conditions (Oklahoma or Texas) = 14.65 psia, 60 °F Basic flow equation is,

$$Q_h = C' \sqrt{h_w p_f} \quad (11\text{-}1)$$

where, $C' = F_b \cdot F_r \cdot Y \cdot F_{pb} \cdot F_{tb} \cdot F_{tf} \cdot F_{gr} \cdot F_{pv} \cdot F_a \cdot F_{am} \cdot F_{w1} \cdot F_{wt} \cdot F_{pw1} \cdot F_{hgm} \cdot F_{hgt}$ (11-2)

1. Base orifice factor, F_b

Fb = 460.80 (GPSA, p. 3-14, F 3-18)
Need flange taps
pipe ID = D = 4.026 in.
orifice diameter = d = 1.500 in.

2. Reynolds number factor, F_r

$F_r = 1 + b/\sqrt{h_w p_f}$

$= 1 + 0.0336/\sqrt{(65)(665.4)} = 1.0002$

Need flange taps
pipe ID = D = 4.026 in.
orifice diameter = d = 1.500 in.
b = 0.0336 (GPSA, 1987, p. 3-6, Fig (3-19)
P_f = 665.4 psia
h_w = 65 in W.C.

3. Expansion factor, Y_1

As defined, ß = d/D = 1.500/4.026 = 0.373
h_w/P_{f1} = 65/(651 + 14.4) = 0.098
From GPSA, p. 3-18, Fig. 3-20

	ß = 0.3	(0.373)	0.4
h_w/p_{f1}	0.0 1.000		1.000
	0.1 0.9989		0.9988
Interpolating	0.098 0.9989	0.9988	0.9988

Need flange taps
static pressure from upstream tap and h_w/P_{f1}

4. Pressure base factor, F_{pb}

Fpb = 14.73/Pb
= 14.73/14.65 = 1.0055
Need P_b base pressure for desired scf.

5. Temperature base factor, F_{tb}

$F_{tb} = (T_b + 460)/(60 + 460)$

$= (60 + 460)/(60 + 460) = 1.0000$

Need T_b = base temperature for desired scf.

6. Flowing temperature factor, F_{tf}

$F_{tf} = \sqrt{(520)/(T_f + 460)}$

$= \sqrt{(520)/(100 + 460)} = 0.9636$

Need T_f = flowing temperature.

7. Specific gravity factor, F_{gr}

$F_{gr} = \sqrt{1/SG} = \sqrt{(1)/(.60)} = 1.2910$

Need SG, gas specific gravity relative to air or gas composition and calculate SG.

8. Supercompressibility factor, F_{pv}

F_{pv} = 1.0401, ANSI/API 2530 (1985), p. 103,
Table D-5 Alternatively,

$F_{pv} = \sqrt{1/z} = \sqrt{1/(.917)} = 1.0443$

Note Z= .917 Fig. 3-17
or GPS A (1987, Fig. 23-8)

Need information to calculate Z; usually T_f, P_f, and gas composition.

9. Orifice thermal expansion factor, F_a

$F_a = 1 + 0.000018 (T_f - T_{meas})$

$= 1 + 0.000018 (100 - 68) = 1.0006$

Need stainless steel, T_{meas}.

10. $F_{am} = \{[62.3663 - (14.4 + 65/27.707)/192.4]/ 62.3663\}^{0.5} = 0.9993$.
11. g = 32.1418 when H = 500 ft and L = 35°
    $F_{w1} = (32.1418/32.174)^{.5} = 0.9995$
12. Assume water in manometer was at 60°F during calibration so $F_{wt} = 1.0000$.
13. $F_{pwl} = 0.995$ same as F_{w1}
14. Bellows meter so omit F_{hgm} and F_{hgt}.

    Therefore (at last!)

    $C' = F_b \bullet F_r \bullet Y_1 \bullet F_{pb} \bullet F_{tb} \bullet F_{tf} \bullet F_{gr} \bullet F_{pv} \bullet F_a \bullet F_{am} \bullet F_{w1} \bullet F_{wt} \bullet F_{wt} \bullet F_{pw1}$

    = (460.80) (1.0002) (.9988) (1.0055) (1.000) (.9636) (1.2910) (1.0443) (1.0006) (.9993) (.9995) (1.0000)(.9995)

    = 600.66.

$Q_h = C' \sqrt{h_w p_f}$

= (600.66) V(65)(665.4)

= 125,100 scfh (14.65 psia, 60°F).
Note $F_b \bullet F_{tf} \bullet F_g = (460.80)(.9636)(1.2910) = 573.23$

The product of $F_b \bullet F_{tf} \bullet F_g$ and C' agree within 5%; and so $F_b \bullet F_{tf} \bullet F_g$ is satisfactory for sizing a meter run.

Orifice Meter Operation

Schepers (1981) describes in detail the inspection and calibration of an orifice meter including the plate, differential bellows, orifice charts, differential and static pens, etc. Table 11-2 summarizes some troubleshooting advice.

Table 11-2 Troubleshooting Orifice Meters (after Pulley, 1988)

Problem	Possible Causes
Low or no Reading	Orifice plate installed backward or oversize Flow blocked upstream Density changes in process media or reference leg Pressure-tap holes and/or piping plugged Bypass valve open or leaking Liquid or gases trapped in piping Block or shut-off valves closed Piping leaks, high-pressure side Housing filled with solids Gas trapped in housing in liquid service Liquid trapped in housing in gas service High-pressure (HP) housing gasket leaks Differential-Pressure unit (DPU) tampered with Loose links or movements in mechanism Pointer loose Mechanism out of calibration Corrosion or dirt in mechanism
Indicates High	Orifice partially restricted or too small Leak in low-pressure side piping Loss of liquid in reference leg (liquid level) Gas trapped in low pressure housing, liquid service Liquid trapped in high pressure housing, gas service Low-pressure (LP) housing gasket leaks Bellows range spring broken, or DPU tampered with Loose links or movements in mechanism Mechanism out of calibration
Erratic Indication	Flow pulsating Liquid trapped in gas piping Gas bubble in liquid piping Vapor generator incorrectly installed Reference leg gassy or liquid vaporizing Obstructed bellows travel Gas trapped in DPU, HP or LP pressure housing Mechanism linkage dragging or dirty Pointer dragging on scale plate

Two common types of charts used are the linear and the square root or L10 (Fig. 11-8). As the name implies, the linear chart is divided into equal sized divisions, usually 100. As is emphasized in Figure 11-8, numbers on the square root chart are merely the square root of linear chart readings.

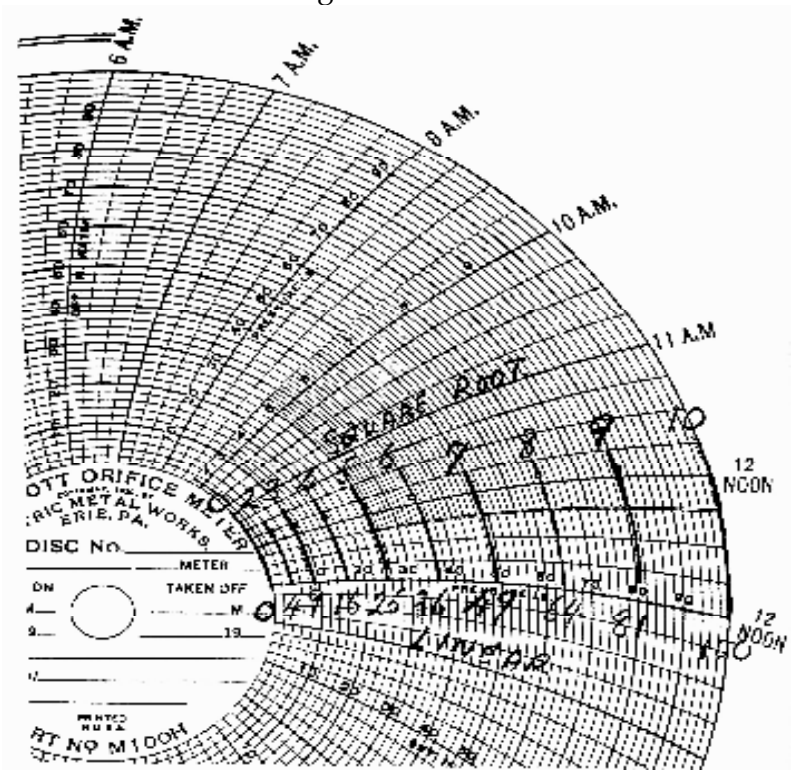


Figure 11-8
Comparison of linear and square root charts.

During calibration the pens should be zeroed correctly. The differential pen is set originally at zero because the meter is calibrated with no flow. However, atmospheric pressure acts on the meter during calibration, and the static pen is set at the current ambient atmospheric pressure. Numerically, the correct static pen "zero" is:

$$\text{Linear: } R_c = (100)(P_{atm})/P_{max} \qquad (11\text{-}14)$$

$$\text{Square Root } R_L = \sqrt{(R_c)} = 10\sqrt{(P_{atm}/P_{max})} \qquad (11\text{-}15)$$

Where

R_c = reading on linear chart
R_L = reading on L-10 chart
P_{atm} = ambient atmospheric pressure (psia) (remember this varies with elevation)
P_{max} = pressure (psia) to cause full scale reading on charts.

Example 11-2. Error caused by incorrect zeroing of static pen. Estimate the error caused by incorrectly zeroing the static pen, i.e., incorrectly setting static pen at zero of chart instead of correctly zeroing pen at 1 atm. Does the orifice meter read high or low?

In the standard orifice equation, $Q_h = C'\sqrt{h_w}\ \sqrt{P_f}$,P_f should be in psia. Incorrect zeroing causes the static element to measure gage pressures: P_f(psia) - P_{atm}.

Therefore

$$Q_h \text{ (correct)} = C'\sqrt{h_w}\ \sqrt{P_f(psia)}$$

$$Q_h \text{ (incorrect)} = C'\sqrt{h_w}\ \sqrt{P_f - P_{atm}}$$

$$Q_h \text{ (incorrect)}/Q_h \text{ (correct)} = \sqrt{h_f(psig)}\ /\ \sqrt{P_{f(Psia)}}$$

The magnitude of the error is now computed for several static pressures. Assume, P_{atm} = 14.2 psia (1000 ft elevation)

Static Pressure (psig)	Pressure (psia)	$\frac{Q_h(incorrect)}{Q_h(correct)}$	Error % Low
35.8	50	0.846	15.4
85.8	100	0.926	6.4
185.8	200	0.964	3.6
485.8	500	0.986	1.4
985.8	1000	0.993	0.7

Note the orifice meter reads low.

Sources of Error

Measurement problems and sources of error are described separately.

Measurement Problems. These include freezing, pulsating flow, slugging, and sour gas. Commonly implemented solutions are shown below (PETEX, 1987):

Freezing
- Install line heaters, heated meter house
- Dry gas, use hydrate inhibitors
- Enlarge meter piping to 0.5 in

Pulsations
- Locate meter away from reciprocating compressor
- Insert capacity, restriction, or filter in line
- Operate at as high as possible h_w

Slugging
- Remove liquids in slug catchers
- Dry gas

Sour gas
- Use 316 stainless static spring
- Use Teflon bearings for differential pen shaft
- Avoid copper piping, mercury manometers

Sources of Error. Figure 11-9 illustrates some common sources of error. Installing the

orifice plate backwards causes the meter to read low by as much as 17 to 30%. A dull plate can easily cause an orifice to read 1 to 2% low. Batchelder (1985) describes these and other sources of error and also documents the importance of witnessing orifice meter calibration and field testing. Burgin (1971) investigated the effects of many abnormal situations on orifice meter readings. Hoch (1983) and Studzinski (1988) also studied orifice plates under abnormal conditions. Table 11-3 summarizes the observations of Batchelder, Burgin, Hoch, and Studzinski. The importance of following all ANSI/API specifications is obvious. These findings also show that orifice meters tend to read low under abnormal conditions.

Current Research Programs

The orifice meter has been the preferred method of gas measurement and custody transfer by pipeline and production companies for over 70 years. The current (1985) ANSI/API Standard 2530 was originally issued by the American Gas Association as AGA Report No. 2 in 1935 and supplemented in 1955 as AGA Report No. 3. It is based primarily on Beitler's (1935) data. The current ANSI/ API 2530 equations were developed by Buckingham and Bean in 1934 (ANSI/API 2530, p. 26). The International Organization for Standardization document, ISO-5167 (1980), adopted the Stolz (1978) equation. Fling and Spencer (1984) estimate that the Stolz equation is based on perhaps 600 flange- and corner-tap data points obtained by Beitler and by Witte in the early 1930s.

Figure 11-10 compares the ANSI/API 2530 and ISO-5167 equations. Even a very small difference in C' becomes very significant when 1 trillion scf of gas are sold.

The American Petroleum Institute (API) and the Gas Processors Association (GPA) have completed development of a new data base of coefficients of discharge for concentric, flanged-tapped, square-edged orifice meters. The program uses three, single-phase, Newtonian fluids (5-centistoke white oil, water, and natural gas) flowing in 2-, 3-, 4-, 6- and 10-in. ID meter tubes, over a pipe Reynolds number range of 150 to 20000000, and yields 21000 new discharge coefficient data points. The data from this project provide a new and many times larger data base than has ever been presented before (Fling, 1986).

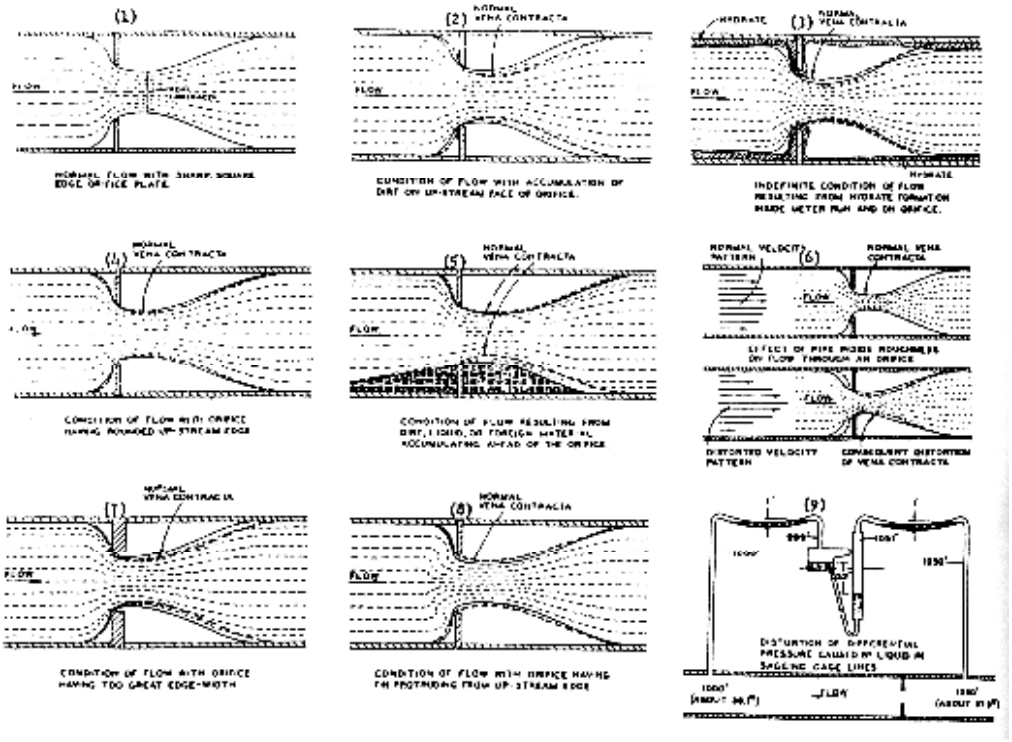


Figure 11-9
Abnormal operating conditions.

Condition	Error	Reference
Plate installed backwards	17% low 14% low up to 30% low	Batchelder (1985) Hoch (1983) Schepers (1981)
Dirt upstream of plate	6% low 11% low	Hoch (1983) Burgin (1971)
Dirt downstream of plate	2% low	Hoch (1983)
Dirt both sides of plate	3% high	Hoch (1983)
Grease and dirt in tube	11% low	Burgin (1971)
Liquid in meter tube	11.3% low	Burgin (1971)
Dirt between pipe and plate on face of plate	23% low	Hoch (1983)
Dirt on upstream face of plate	27% low 0-4.7% low	Hoch (1983) Burgin (1971)
Valve lubricant on upstream face	0-4. 7% low	Burgin (1971)
Valve lubricant on downstream face	1% low to 3% high	Burgin (1971)
Rounded and scratched place	11% low	Hoch (1983)
Dull plate	.5% low	Burgin (1971)
Upstream edge bevelled	2-13.5% low	Burgin (1971)
Upstream edge rounded	0-9.3% low	Burgin (1971)
Warped plate	2% low to 10% high	Burgin (1971)
Methanol injection upstream	6% low	Batchelder (1985)
Static pressure 10 psi low at 221 psig and flow temperature 7°F high at 75°F	3% low	Batchelder (1985)

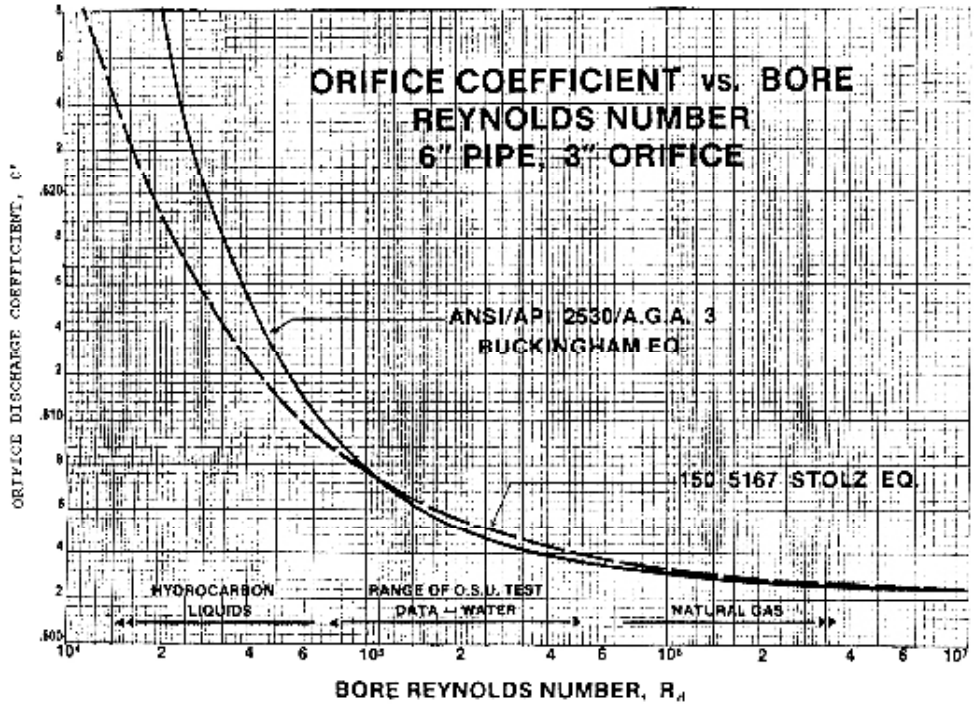


Figure 11-10
Comparison of ANSI/API 2530 and ISO 5167 (Hoglund, 1984).

In the European work, orifice plates of six different diameters (51 mm-191 mm) and two thicknesses were tested at six laboratories in 100- and 250-mm pipes using water, natural gas, and air over a Reynolds number range of 5×10^3 to 3.5×10^7. Over 4000 data points were obtained for flange and D and D/2 taps (CEC, 1985; CEC, 1987).

These 25000 data points have been analyzed, and several new equations for the orifice discharge coefficient have been suggested. Hopefully, agreement will soon be reached on a new and improved equation that will replace the Buckingham and/or Stoltz equations.

A full-scale gas metering facility—the Karsto Metering & Technology Laboratory, or K-Lab—has been built in Stavager, Norway (Bosio, Wilcox, and Sembsmoen, 1988).

The project is a joint venture between Statoil and Total. The facility uses sonic nozzles as reference flowmeters and a gravimetric system for the primary calibration of the sonic nozzles. The facility operates as a closed loop, the maximum volume flowrate will be 460 million scfd; and the operating pressures in the working sections range from 145 to 2262 psia. The objective is to calibrate gas-flow meters in pipes with diameters of 4 to 24 in. with an uncertainty in mass flow of less than 0.25%.

The Gas Research Institute recently initiated the Metering Research Facility program with Southwest Research Institute. In 1987 an extensive industry survey was conducted to identify research needs, the priority of needs, and the expected benefits. This survey resulted in the following priority rating for testing meter types:

1. Turbine
2. Orifice
3. Vortex
4. Positive displacement.

The priority rating for research topics is:

1. Upstream lengths
2. Pulsation effects
3. Swirl.

See McKee (1989) for more details.

TURBINE METERS

Today turbine meters are available for gas measurement in sizes and working pressures ranging from 0.24 to 3.36 MMscfd and 175 to 1440 psig. The construction of turbine meters is presented first and then meter performance is reviewed by discussing the validity of two basic assumptions: first, that the rotor rotation varies linearly with the average fluid velocity, and second, that the volumetric flowrate is proportional to the average fluid velocity. Finally, meter sizing and accuracy are summarized and a troubleshooting checklist is presented.

Construction of Turbine Meters

As shown in Figure 11-11, a turbine meter consists of (Baker and Kalivoda, 1977):

1. Housing—a flanged pipe spool .25 to 24 in. diameter with 275-6000 psig working pressure and -20 to +500°F standard design temperature. Construction material is usually carbon steel or stainless steel for corrosive environments or low temperature applications.
2. Upstream and Downstream Hangers (or stators, supports)—which center and support the rotor and axially clamp the rotor thrust bearings. These hangers contain the thrust washers, provide passages for hydraulic thrust balancing of the rotor, and include blades for straightening the flow.
3. A Rotor—with stainless steel blades supported by tungsten carbide journal bearings and thrust washers. These tungsten carbide bearings are highly polished and have a small bearing surface to minimize drag. In modern designs the rotor "floats" between the upstream and downstream cones on a thin film of liquid that flows between the bearing and the shaft. The rotor is thrust upstream by the pressure difference across the rotor blades and downstream by the flow impinging on the outer rim of the rotor hub which is, by design, not shielded by the upstream cone.

4. A Variable Reluctance Pick-up Coil—which detects the rotational speed of the rotor by monitoring changes in magnetic flux passing through the coil bobbin. The rotor must have regularly spaced paramagnetic material at the periphery for the variable reluctance pickup to work (Daniel, 1983).

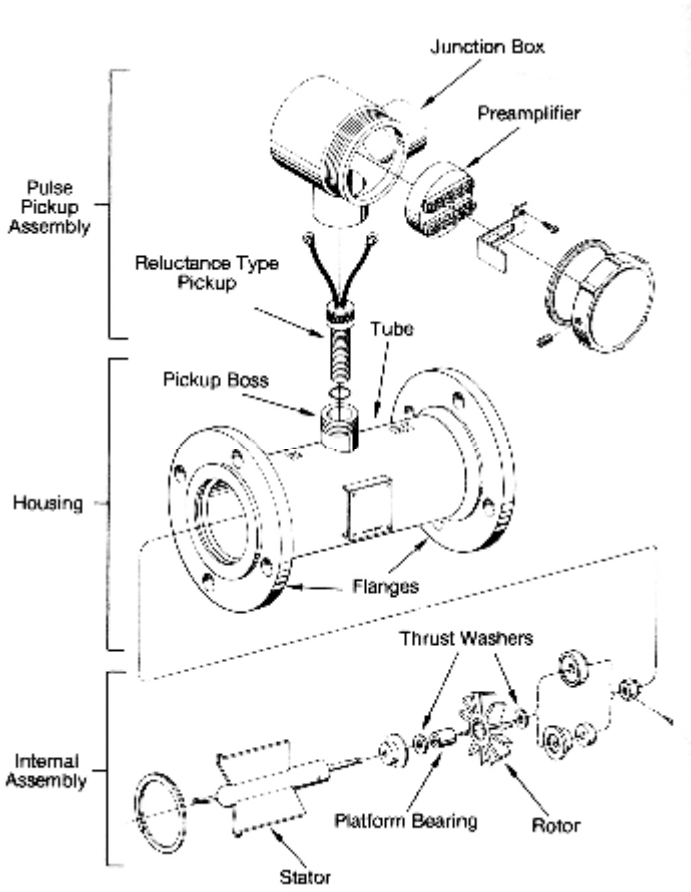


Figure 11-11
Turbine meter construction (Laird, 1988).

Gas Turbine Meters

In principle, gas turbine meters are the same as for liquids, with a few important differences (Fig. 11-12). Since the driving torque is proportional to the density of the flowing gas, this torque is much lower than for liquids. The rotor speed is therefore maintained high by operating at high pipeline velocities and by having a high ratio of center body diameter to pipe diameter. A nose cone or flow deflector forces the gas to flow through an annulus having an open area approximately one-third of the open pipe area, thus providing more driving torque. The nose cone also absorbs most of the flow stream thrust that otherwise might damage the rotor bearings.

The rotor spins at similar speeds to those for liquids, and hence smaller blade angles are used (10 degrees) compared to liquids (35 degrees). The rotor blades are often helical rather than flat and are machined or molded as an integral part of the hub to improve strength. Because light weight improves rotor performance and bearing life, high- strength, impact-resistant plastic or alumina is normally used.

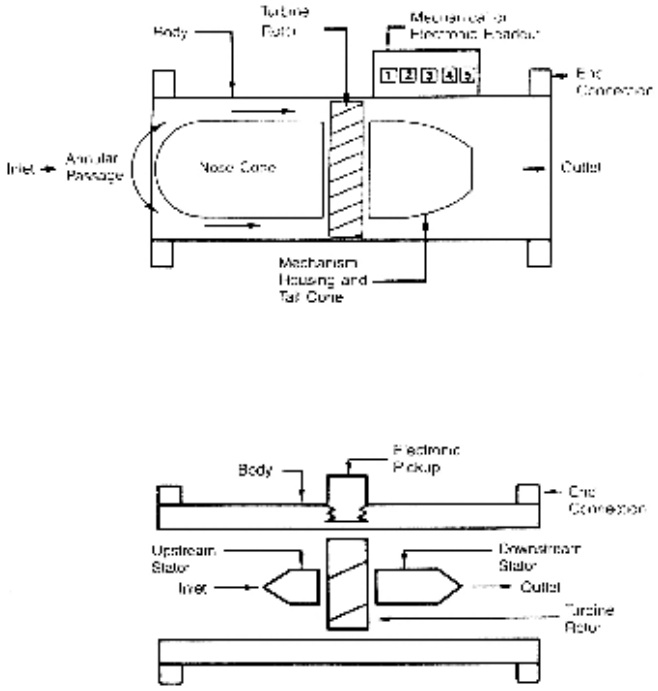


Figure 11-12
Gas turbine meter (AGA, 1985).

Bearings are usually of the ball race type and small relative to the meter partly to reduce frictional drag and partly due to the high rotational speed. The nose cone usually shields the bearings from liquids, dirt, and grit entrained in the following gas. Bearings must be lubricated either permanently or periodically during operation. Most meters use a wick or felt to add lubricating oil, even when the meter is pressurized. Excess oil in the bearings flushes out any dirt and eventually evaporates into the gas stream (Allen, 1989).

There are two distinct methods of measuring rotor rotation—mechanical and electro-magnetic. Mechanical designs use gear trains connected to a counter clock. There are four types of electro-magnetic sensors—induction pickup coil, reluctance pickup coil, modulated carrier pickup coil, and light emitting diode sensor (Allen, 1989).

Many flanged meters are designed to have the internal working parts contained in a module assembly that is easily accessible for maintenance or replacement. All rotating parts which are likely to wear or erode are part of this module.

Due to the large contraction in area, gas meters tend to be less influenced by inlet conditions than liquid meters. Nevertheless, some 20 pipe diameters are recommended as the minimum inlet length even though the flow standards that are appearing suggest 10 diameters as a minimum length.

Calculation of Gas Flow Rate

The recommended method for calibrating turbine meters is the transfer test in which a volume of gas (or air) is passed through the meter in series with a "master meter," or a critical-flow or a bell prover, or sonic nozzle. Standard factory curves are established. Tests at 100 psig are adequate to establish meter performance at elevated pressures (Kemperman, 1986).

The basic volumetric flow rate equation is

$$Q = (MF)\ (\text{Dial Diff})\ (C_t)\ (C_p)\ (C_z) \qquad (11\text{-}16)$$

where

Q = scf at 14.73 psia and 60°F
MF = meter factor (normally set at 1)
Dial Diff = difference between ending and beginning meter dial readings
C_t = temperature correction factor
= 520/(460 + flowing temp °F)
C_p = pressure correction factor
= flow pressure (psia)/(14.73)
C_z = compressibility factor correction
= (1)/(Z at flowing T and P).

In most cases the manufacturer lists the capacity and pressure drop for a 0.6 SG gas. These curves can be converted for other conditions as follows:

$$\Delta P_f = \Delta P_r \left(\frac{\rho_f}{\rho_r}\right)\left(\frac{Q_f}{Q_r}\right)^2 \qquad (11\text{-}17)$$

$$Q_{x,max} = Q_{.6,max} \left(\frac{P_f}{P_r}\right)\left(\frac{.6}{SG}\right)^{0.5} \qquad (11\text{-}18)$$

where

ΔP = pressure drop across meter
ρ = gas density
Q = flow scf
P = static pressure psia
SG = gas gravity

Subscripts

f = current case
r = reference or calibration with
G = .6
x = refers to x gravity gas.

Operation

To avoid problems and prolong meter life (Kemperman, 1986), do the following:

1. Flush bearings regularly with high-grade, low viscosity instrument oil.
2. If necessary, install an upstream strainer and/or a filter separator to remove solid particles and liquids from the gas.
3. Protect meter from weather by a suitable cabinet.
4. Operate meter within its range.

Meter testing, onsite proving, operating problems, and troubleshooting are now summarized.

Table 11-4
"Trouble-Shooting Guide" (Furness, 1984)

Condition	Cause	Correction
No output pulses from amplifier module.	1. Input voltage to amplifier below minimum required for operation. 2. Damaged amplifier module. 3. Receiver unit not operating. 4. Turbo-meter rotor not turning.	1. Replace pick-off. 2. Replace amplifier module. 3. Refer to instruction manual on defective unit. 4. Trouble shoot rotor.
Turbo-meter rotor not turning.	1. Defective rotor shaft bearings. 2. Rotor damaged by foreign material passing through meter.	1. Replace bearings. 2. Send rotor to factory for replacement or repair.
No mechanical output (mechanical type meters such as Brooks, Rockwell, etc.)	1. Sheared coupling. 2. Rotor not turning. 3. Bevel gears worn out. 4. Gears in register cup worn out. 5. Defective accessory equipment. 6. Shaft not aligned with coupling. 7. Driven magnet slipping on shaft. 8. Broken shaft.	1. Replace coupling. 2. Trouble shoot rotor. 3. Replace bevel gears. 4. Replace defective gears in register cup. 5. Refer to instruction manual on defective unit. 6. Align shaft with coupling. 7. Tighten set screw in magnet. 8. Replace the broken
Inaccurate readout.	1. Foreign material on rotor blades. 2. Rotor blades bent. 3. Defective accessory equipment. 4. Defective magnet.	1. Check and clean blades. 2. Send to factory for replacement or repair. 3. Troubleshoot equipment. 4. Replace magnet.

Meter Testing. The "spin test" is a simple, fast, and reliable way to detect any increased friction. A spin test consists of spinning the meter rotor either by hand or with an air jet and then measuring how long it takes for the rotor to coast to a complete stop (spin time).

Onsite Proving. Beeson (1988) describes in detail how to prove a turbine meter using a sonic nozzle. An accuracy of 0.5% is claimed.

Operating Problems. Overloaded meters and/or poor gas quality are the major problems (Lofton, 1988). Do not operate the turbine meter continuously at over 70% of rated capacity. Entrained liquids—water and/or hydrocarbons—tend to wash the lubricants out of the ball bearings. Frequent lubrication and protection of the ball bearings is recommended (Lofton, 1988).

Troubleshooting. See Table 11-4. Consult AGA (1985) for detailed specifications and recommendations for gas turbine meter construction, installation, operation, performance characteristics, volumetric flow measurement, calibration, and field checks.

VORTEX METERS

When a fluid impinges on a bluff or nonstreamlined body, it splits into two paths. The resulting instability of the fluid flow field causes alternating vortices to shed from each side of the bluff body at a frequency proportional to the incident fluid velocity (Fig. 11-13). This phenomenon is readily visible when a flag waves in the breeze. The flag pole serves as the bluff obstruction and generates vortices that cause the flag to wave. Such vortex shedding can produce sound as when a wire vibrates and "sings" in the wind. Note that the vortices are formed alternatively, first off one side of the body and then off the other side and are 180 degrees out of phase. This so-called von Karman vortex street results in areas of alternating high and low pressure. Vortex meters usually use piezoelectric crystals that act as force-to-charge transducers to detect these pressure fluctuations. Figures 11-14 and 11-15 show Foxboro Co. vortex flowmeters in which the bluff, or vortex-shedding, body also houses the piezoelectric sensor. Fisher Controls uses a dual bluff body design (Fig. 11-13) to amplify the vortex signal, while Foxboro locates the sensor in a quiet fluid zone (Fig. 11-15). Both approaches reduce background noise.

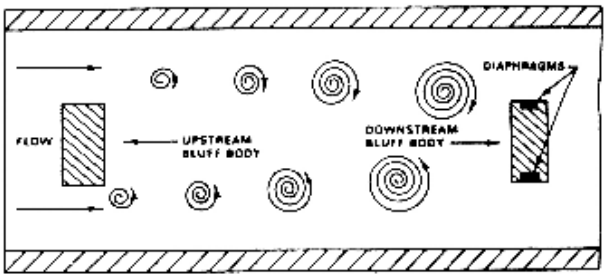


Figure 11-13
Vortex formation (Todd, Fisher Controls, 1988).

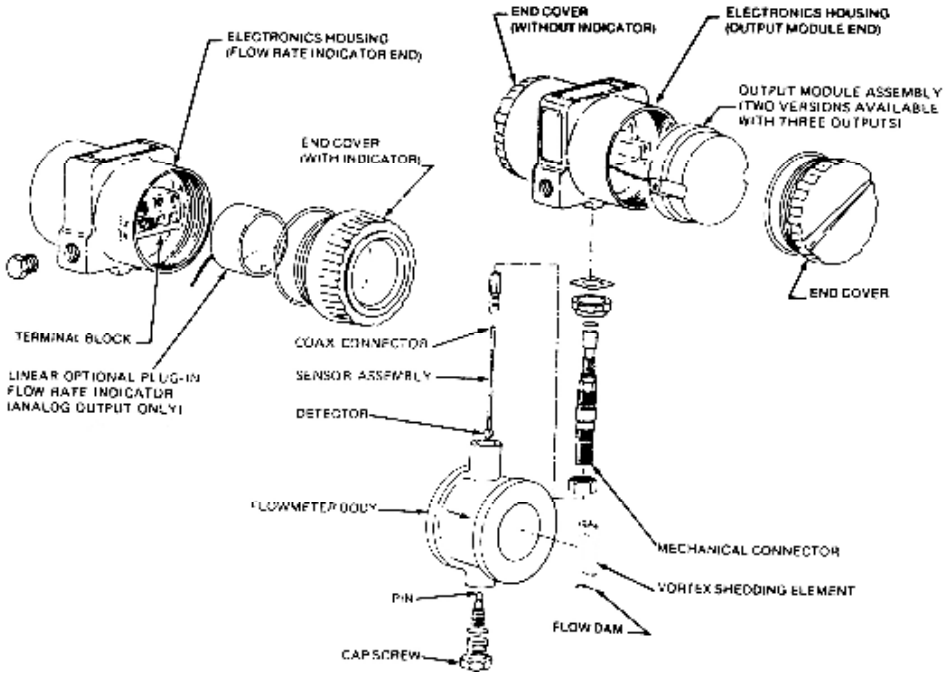


Figure 11-14
Vortex flowmeter, wafer style (Echeverria, 1985).

Gas Sizing

The Strouhal number, St, relates the frequency of vortex formation, f, to the fluid velocity, v.

$$St = f\, d_b / v \qquad (11\text{-}19)$$

where d_b = width of bluff body.

Figure 11-16 shows that the Strouhal number remains constant when the pipe Reynolds number, $D\, v\, \rho/\mu$, ranges from 104 to 106. Vortex shedding does not occur at Re less than 3000.

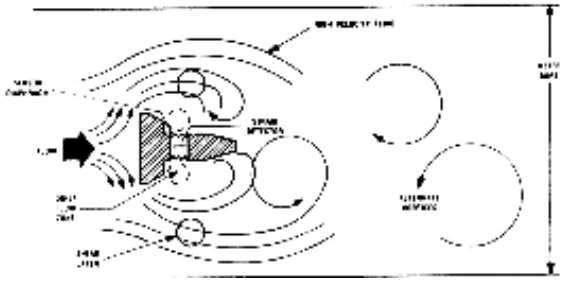


Figure 11-15
Vortex meter, Foxboro Co. (Echeverria, 1985).

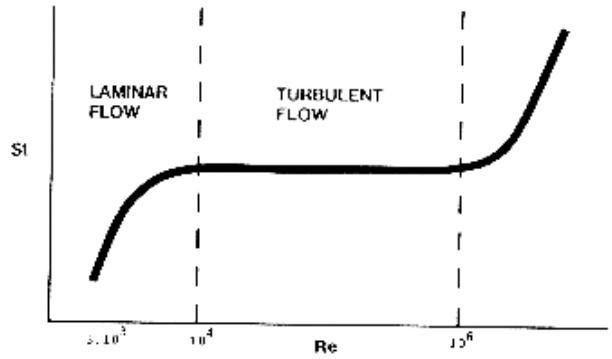


Figure 11-16
St No. vs. Re No. (Todd, 1988).

While the vortex-shedding frequency varies linearly with the fluid velocity over a very wide range, in practice the measuring range is limited by the ability of the piezoelectric sensor and electronics to detect and amplify the pressure pulses. Mathematically, the differential pressure is given by

$$\Delta P = C_{vm}\, \rho_f\, \bar{v}^{2} \qquad (11\text{-}20)$$

where

ΔP = differential pressure, psi
ρ_f = fluid density (lb/ft^3)
v = average fluid velocity, ft/s
C_{vm} = proportionality constant.

Rearranging Equation 11-20:

$$\bar{v} = \sqrt{\Delta P / C_{VM}\, \rho_f} \qquad (11\text{-}21)$$

$$\bar{v}_{\min} = a/\sqrt{\rho_f} \text{ and } \bar{v}_{\max} = b/\sqrt{\rho_f} \qquad (11\text{-}22)$$

where

$\bar{v}_{\min}$ = minimum detectable velocity
$\bar{v}_{\max}$ = maximum velocity that will not damage sensor
a,b = constants for a specific meter.

The limit velocities and C_{vm} are determined empirically for each specific meter design. For the Foxboro E83 vortex flowmeter (Echeverria, 1985):

a = 5 for gases, a = 10 for liquids

b = 158 for both gases and liquids

To size a vortex meter, compute $\bar{v}_{max}$ using either the frequency limitation (max Re for which St is constant) or the sensor and electronics limitation, whichever is lower. Then:

$$Q_{vm}=A_b \bar{v} \qquad (11\text{-}23)$$

where

Q_{vm} = actual flowrate (ft^3/s)

Ab = meter bore area (ft^2).

The smallest bore area provides the cheapest flowmeter with the greatest rangeability.

ULTRASONIC FLOWMETERS

Ultrasonic flowmeters can be divided into four basic types: time of flight (TOP), Doppler, cross-correlation, and swept-beam (Cascetta and Vigo, 1988). The best known types are the convenient but not very accurate Doppler and the more accurate and more expensive TOF. Only the TOF type is discussed further because this is the most popular oilfield type.

The TOF ultrasonic flowmeter consists of two piezoelectric sensors located 180 degrees apart and separated by an axial distance, L, as shown in Figure 11-17. When a voltage pulse is applied to sensor A, it changes its mechanical dimension—alternately expanding and contracting—and so generates an ultrasonic energy pulse. This pulse travels at sonic velocity through the fluid and is received by sensor B. The fluid velocity, v_f, is proportional to the difference between the transit times for the pulse to travel upstream, t_{BA} (from sensor B to sensor A), and downstream, t_{AB} (A to B). Velocity is given by (Kyser, 1988)

$$v_f = D(t_{BA} - t_{AB})/t^2 \sin 2\theta_f \qquad (11\text{-}24)$$

$$\text{where } t = D/\cos \theta_f \, v_{son} \qquad (11\text{-}25)$$

v_{son} = sonic velocity in fluid.

Scelzo and Munk (1987) report a three-year field test of a TOF ultrasonic flowmeter for natural gas pipelines 6 to 30 inches in diameter. Their data confirms a 50:1 turn-down ratio and a ±2% accuracy. Ultrasonic flowmeters also are being used to measure flare gas in the North Sea (Mylvaganam, 1988).

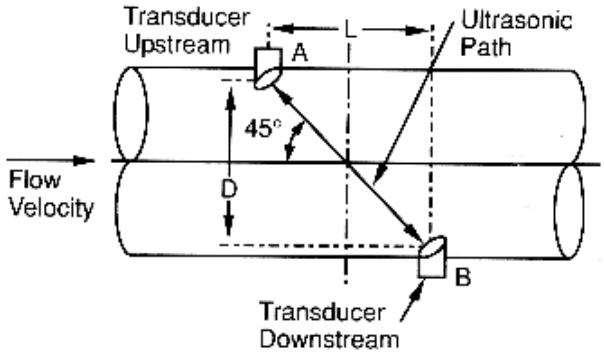


Figure 11-17
Across-the-pipe ultrasonic flow meter geometry (Scelzo and Munk, 1987).

SELECTION CRITERIA

The relative advantages and disadvantages of orifice, turbine, vortex, and ultrasonic meters are reviewed first. Then uncertainty and repeatability are reviewed.

Relative Advantages and Disadvantages
Orifice Plates.

Advantages: Accepted as international standards
Simple to use
No moving parts
Do not require frequent maintenance/ calibration

Disadvantages: Square root head/flow relationship
High permanent pressure drop
Limited rangeability - 4:1
Tends to read low under abnormal conditions

Turbine Meters.
Advantages: Very good repeatability—± 0.1%
High rangeability—10:1
Low pressure drop

Disadvantages: Moving parts subject to wear
Bearings damaged by overspeeding, corrosion
Rather expensive
Dirty fluids must be filtered

Vortex Meters.

Advantages: Good rangeability-30:1
No moving parts
Good linearity over operating range
Disadvantages: Not suitable for dirty/abrasive fluids
Appreciable pressure drop
At lowest velocity Re > 10^4

TOP Ultrasonic.

Advantages: No intrusion into pipe
Wide rangeability - 50:1
Easy to install ("clamp on")
Cost almost independent of size
Disadvantages: Periodic calibration required
Accuracy not better than ± 2% FS
Gases must be clean.

Uncertainty and Repeatability

Much confusion exists in some of the terminology such as accuracy, error, uncertainty, and repeatability.

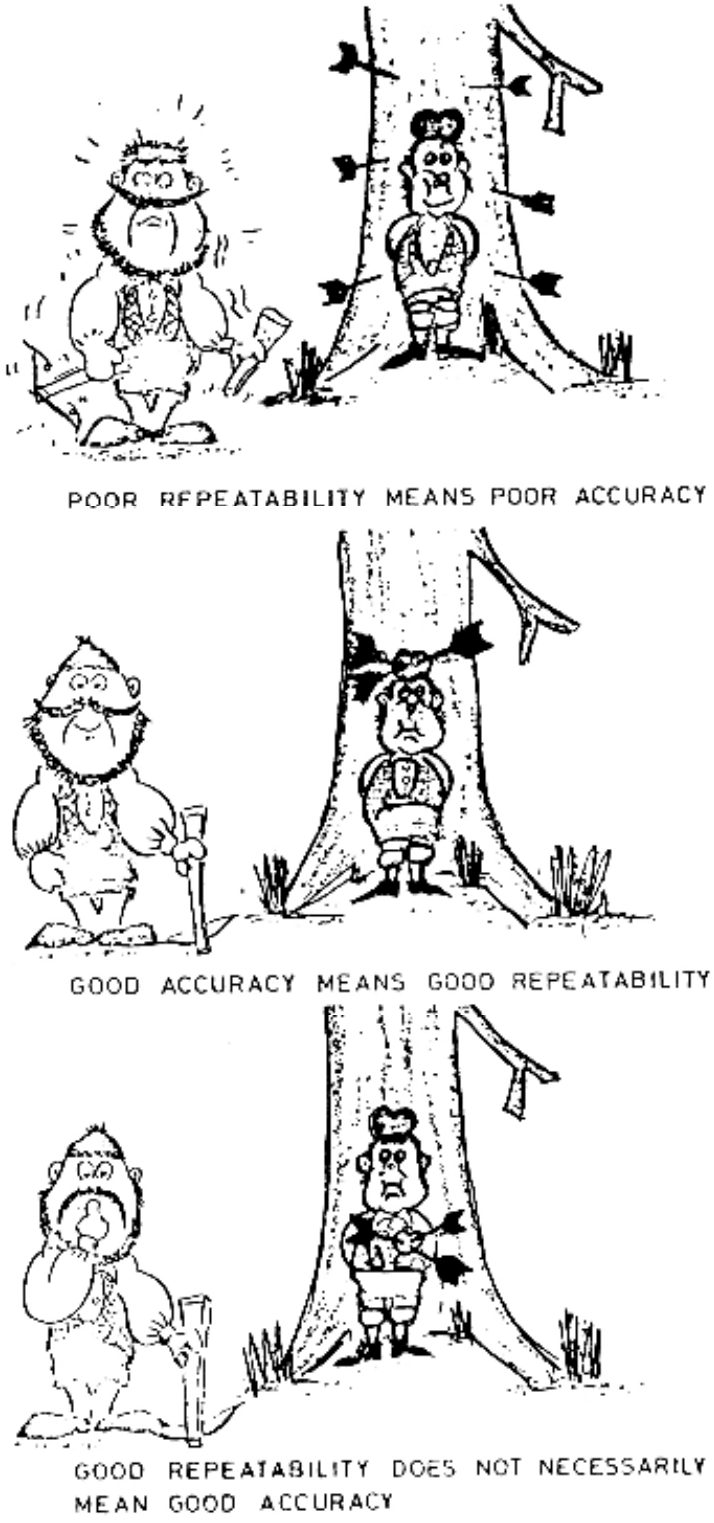

POOR REPEATABILITY MEANS POOR ACCURACY
GOOD ACCURACY MEANS GOOD REPEATABILITY
GOOD REPEATABILITY DOES NOT NECESSARILY
MEAN GOOD ACCURACY

Figure 11-18
illustrates the difference between accuracy and repeatability.(Hydril)

Error is the difference between the measured and true values, and is nearly always unknown unless the true value is provided, e.g., a standard sample of known, made-up composition.

Uncertainty is half the range within which the true value is expected to lie with a stated probability. Unlike error, uncertainty can be calculated using proper statistics, but the uncertainty cannot be separated from the probability or confidence level with which it is associated.

Repeatability or precision can detect spurious and random errors but fail to measure constant systematic errors or bias.

Variable systematic errors such as progressive wear of a turbine meter's bearings or progressive dulling of an orifice plate are also hard to detect.

Norman and Jepson (1986) estimated the measurement of orifice and turbine flowmeters as summarized in Tables 11-5 and 11-6. ANSI/API 2530 assumes that all the involved variables are independent of each other—by far the simplest case to compute. ISO 5168 (1978) accounts for the dependence of the calculated compressibility factor on the measured pressure, P_f; temperature, T_f; and gas gravity, SG.

Table 11-5
Comparison of ISO and AGA3 Uncertainties for Orifice Plate Meters (Norman and Jepson, 1986)

Variable	Percent Uncertainty of Variable (±)	Effect factor	
		Exponent	Square
F_b Basic Orifice Factor	0.5	1 (1)	0.25 (0.25)
F_r Reynolds Number Factor	0.1	1 (1)	0.01 (0.01)
Y Expansion Factor	0.1	1 (1)	0.01 (0.01)
T_f Temperature	0.4	1.84/2 (½)	0.1354 (0.04)
G_r Relative Density	0.3	0.68/2 (½)	0.0104 (0.0225)
Z_{fl} Compressibility	0.25	½ (½)	0.0156 (0.0156)
h_w Differential Pressure	0.5	½ (½)	0.0625 (0.0625)
P_f Absolute Static Pressure	0.5	1.21/2 (½)	0.0915 (0.0625)
Sum of Squares			0.5854 (0.4731)
Overall Uncertainty (Square root of Sum)			±0.765%(±0.688%)

(AGA3 values in brackets)

Table 11-6 Example of Uncertainty Calculations for a Turbine Meter (Norman and Jepson, 1986)

Variable	Percent Uncertainty of Variable ±	Effect factor	
		Exponent	Square
Q_{act} Basic Turbine Factor	0.5	1 (1)	0.25 (0.25)
T_f Temperature	0.4	1.84 (1)	0.5417 (0.16)
G_r Relative Density	0.3	0.32 (0)	0.0092 (0)
Z_{fl} Compressibility	0.25	1 (1)	0.0625 (0.0625)
P_f Absolute Static Pressure	0.5	1.21 (1)	0.3660 (0.25)
Sum of Squares			1.2294 (0.7225)
Overall Uncertainty (Square Root of Sum)			±1.11% (±0.85%

(Assuming variables independent in brackets)
Tables 11-5 and 11-6 readily support two conclusions:

1. The orifice plate appears to be more "accurate" than the turbine meter.
2. ISO 5168 estimates higher uncertainties than ANSI/API 2530.

While ANSI/API 2530 provides definitive specifications and recommendations for the primary element (orifice plate and meter tube) it does not address secondary elements used to measure static pressure, temperature, gravity, gas composition. Such elements as pressure and temperature recorders or integrators are just as important as the primary element. Poor equipment design, improper installation, and bad operating practices can, of course, cause major error— for example, installing the orifice plate backwards causes the meter to read low by 14 to 30%! Manual integration of the orifice charts will introduce additional uncertainty that can vary from 0.1 to 1.4% (Jones, 1986).

Batchelder (1985), Scheppers (1981), and Woods (1989) summarize correct procedures for field testing and witnessing orifice meter calibration. Ting et al. (1989) discuss and compare chart recorders and flow computers (dedicated microprocessor-based measurement and computation devices). Ting et al. also examined the thermal stability, hysteresis, line-pressure effect, and long-term performance of a chart recorder and a flow computer. They drew the following conclusions:

1. Both the chart recorder and flow computer met their specifications at room temperature and low pressure.
2. Long-term flow deviations due to time and P and T variations were within 0.5% for the flow computer and up to 1.9% for the chart recorder.
3. All differential pressure transducers exhibited significant temperature effects at low temperature.

The authors conclude that "flow computer bests chart recorder."
Birkhead (1986) reports a comparison of a single 4—in. turbine meter in series with an existing dual 4—in. orifice meter station offshore. After measuring 1.2 billion scf the turbine results were 0.05% higher than the orifice measurements.

Nearly always, gas flow meters are designed, analyzed, and tested assuming steady-state flow. However, very often these meters are exposed to flow pulsations caused by reciprocating compressors, pressure regulators, and other disturbances. McKee (1987) compared pulsation effects on various gas meters. Pulsations causing between 10 and 35% velocity modulation are not uncommon at field meter sites, and this level starts to cause noticeable results. Very severe pulsations—75 to 100% velocity modulation—cause large meter errors—as much as 20% high for orifice, 50% for turbine, 30% low for tee-shaped vortex, and 50% high for wedge-shaped vortex meters. Rotating vane positive displacement meter shows 1-5% error which just exceeds the uncertainty limit in the overall experiment (McKee, 1987). McKee concludes that orifice meters are less affected by pulsation than turbine and vortex meters.

GAS ODORIZATION

Methane and sweetened natural gas are odorless. Therefore, a leak could easily go unnoticed, thus causing an explosion hazard at concentrations between the lower (5 mol percent in air) and higher (15 mol percent) flammability limits. Suffocation occurs when the natural gas concentration in air exceeds 50 volume percent.

In the U.S., Federal law (CFR 49, Paragraph 192.625) requires odorization of natural gas and its products before transportation in pipelines. Basically, the law states that: "A combustible gas in a distribution line must contain a natural odorant or be odorized so that at a concentration in air of one-fifth of the lower explosive limit, the gas is readily detectable by a person with a normal sense of smell."

Some State regulations are more severe and require detection at one-tenth of the lower explosion limit.

There are some exceptions. Gas used in the field—heater treaters, gas engines, gas lift, cycling, repressuring—need not be odorized. Gas pipelined directly to gas plants and refineries also can be exempted. See the Federal regulations for the specific exemptions.

LP-Gas Odorants

In 1931 the Bureau of Mines studied many potential odorants for natural gas (Fieldner et al., 1931). This very comprehensive study formed the basis for establishing the current practice of adding 1 Ib of ethyl mercaptan per 10,000 gallons of LP-gas. Today, LP-gas is usually odorized when it is distributed to wholesalers or retailers who, in turn, sell to many customers. Using multiple odorants would risk chemical interactions. Ethyl mercaptan was probably chosen because its high vapor pressure helps it vaporize as the LP-gas changes from liquid to vapor in the LP-gas tank.

Ethyl mercaptan is not suitable as a natural gas odorant because of its lower stability. Ethyl mercaptan can ' 'FADE'' or be oxidized:-

$$3Fe2_2O_3 + 2C_2H_5SH = C_2H_5\text{-}S\text{-}S\text{-}C_2H_5 + 2Fe_3O_4 + H_2O.$$

The produced disulfide does not have as strong an odor. Fading notwithstanding, ethyl mercaptan is almost universally used for LP-gas which is a "clean" liquid hydrocarbon mixture.

Natural Gas Odorants

Gas pipelines can and do contain water, rust, pipline chemicals and other impurities. This more reactive environment dictates a less reactive odorant. Natural gas is usually odorized just prior to its distribution to ultimate customers. Multiple odorants can therefore be used.
Natural gas odorants are usually blends of two or more sulfur-containing hydrocarbons from three basic groups: mercaptans, alkyl sulfides, and cyclic sulfides. Johnson (1985) describes in detail the more-commonly-used component chemicals and the most popular "blends."

Odorants such as tertiary butyl mercaptan (TBM), isopropyl mercaptan (IPM), dimethyl sulfide (DMS), and tetrahydrothiophene (THT) (also called thiophane) (THT) can be readily detected at 1 ppb by a person with average sense of smell. Addition of 0.5 to 0.75 Ib odorant per MMscfis adequate to give natural gas its "gassy" smell. When a new pipeline is first used, 2 Ib odorant per MMscf are added to allow for adsorption of the odorant on the pipeline wall. This adsorption provides a safety factor in the event that odorization equipment fails temporarily.
LP-gas is odorized by adding 1 lb ethyl mercaptan per 10000 gal liquid LP-gas.

WARNING. Be extremely careful when handling liquid odorants—any spill will create a very obnoxious and long-lasting stink.

Odorizers

Odorants are added by two basic methods:
1. Evaporating the liquid odorant and mixing the resulting vapor with the natural gas,
2. Injecting liquid odorant directly into the gas pipeline.

In the vaporization approach, a bypass stream of natural gas absorbs odorant by flowing next to a saturated wick or over the surface of the liquid odorant. Then the resulting gas-odorant mixture is returned to the pipeline. This type is low cost, ideally suited for small flows, and requires minimum maintenance—periodic filling of the odorant storage chamber (Katuran,1989).

There are three types of liquid injection odorizers: drip, meter-driven pump, and positive displacement (PD) pump. The drip type is often used to provide temporary supplemental odorization in new lines. The meter-driven type uses the pressure drop across an orifice at the full gas stream pressure to drive a meter located in the odorizer compartment. In turn, this meter operates a scoop that dips liquid odorant from a vessel and pours it into a tube leading to the gas stream. Odorant is added proportionally because meter rpm varies directly with the main gas flow. Either diaphram or piston-type PD pumps are used for high gas flows. Control

instrumentation is available to add odorant in direct proportion to the gas flow (Katuran, 1989).

Contact the U.S. Department of Transportation for detailed instructions on operation, maintenance, sampling, and record keeping.

Review Questions

1. What type of flow meter is used predominantly for custody transfer of natural gas?
2. Name two other types of meters that are finding increasing use.
3. What type of pressure tap is used most frequently on orifice meters?
4. What is the chief advantage of a senior orifice fitting? What is its main disadvantage?
5. Are orifice meters calibrated in the field?
6. Are orifice plates calibrated in the field?
7. List the precautions required for accurate orifice metering.
8. The orifice flow constant, C', is, in general the product of 15 factors. List these factors.
   Which of these factors differ significantly from 1 .0?
9. Describe the correct zeroing of an orifice meter.
10. Estimate the size of the error caused by incorrectly zeroing the static pen at 0 psia instead of at 1 atm.
11. List the major problems encountered in orifice measurement of natural gas.
    How are these problems handled?
12. List some of the common situations that produce abnormal conditions in orifice metering.
13. Do abnormal conditions tend to cause the orifice meter to read high or low?
    Do abnormal conditions tend to benefit the seller or the buyer?
14. How much error results if an orifice plate is installed backwards? Who is hurt — the buyer or the seller?
15. Does a dull orifice plate read high or low? Estimate the magnitude of the error.
16. How can orifice plate sharpness be measured?
17. What error would you expect from a dirty orifice plate?
18. Describe a gas turbine meter.
19. What precautions are recommended to avoid operating problems with a gas turbine meter in order to prolong meter life?
20. What phenomenon causes a flag to wave in the breeze?
21. Describe a gas vortex meter.
22. Describe an ultrasonic gas meter.
23. Compare orifice, turbine, vortex, and ultrasonic gas meters. List their advantages and disadvantages.
24. Define accuracy, uncertainty, and repeatability.
25. Which are more accurate—orifice or turbine meters?
26. Why is natural gas odorized?

What types of compounds are used to odorize natural gas?
How are odorants added?
How much odorant is usually added?

Problems

1. Natural gas is flowing in a pipeline at a bore Reynolds number of 10^6. One billion (10^9) scf per day are sold at say $1.50/Mscf.
   Estimate the difference in daily revenue ($/day) if ISO 5167 is used instead of ANSI/API 2530.
   Use Figure 11-10.

2. A small lease gathers low pressure gas, compresses it, and sells it to a high-pressure pipeline. Typically, 1 MMscfd are gathered and purchased at pressures ranging from 50 to 100 psig and sold at about 800 psig.
   The proprietor loudly proclaims "A zero is a zero is a zero." He insists on setting both the differential and static pens at the bottom line of the chart when he calibrates his purchase orifice meters. Can you offer a financial reason for the operator's reluctance to follow correct calibration procedures? The low-pressure, purchase orifice meters have static pressure springs with ranges of 50 or 100 psia. The pipeline custody transfer meter uses a 1000 psia static pressure spring.

3. An orifice meter run has the following characteristics:
   Meter Pipe 6 in. schedule 40 (6.065 in. ID)
   flange taps
   static pressure measured from upstream tap
   Orifice stainless steel
   Plate 2.00 inch at 20°C (68°F)
   Recorder 100 in. WC differential
   1000 psia static spring
   Readings atmospheric pressure = 14.2 psia (1000 ft elevation)
   flowing temperature = 90°F
   gas gravity = 0.60
   static pressure = 703 psig
   differential = 59 in. (bellows)
   Base Conditions Oklahoma or Texas 14.65 psia, 60°F
   Calculate the Meter Coefficient and Gas Flow rate. Estimate the error caused by incorrectly zeroing static pen, i.e., incorrectly zeroing static pen at zero of chart instead of correctly zeroing pen at 1 atm.

Nomenclature

A,B = locations of sensor in ultrasonic meter (Fig.11-17)
A_b = vortex meter bore area (ft^2)
a = constant for v_{min} (Eq. 11-22)
b = constant for v_{max} (Eq. 11-22)
C' = orifice flow constant (Eqs. 11-1, 11-2)
C_p = turbine meter pressure correction (Eq. 11-16)
C_t = turbine meter temperature correction (Eq. 11-16)
C_{vm} = proportionality constant (Eq. 11-20)
C_z = turbine meter compressibility factor (Eq. 11-16)
D = pipe inside diameter (in. or ft)
DP = differential pressure
d = orifice diameter (in. or ft)
d_b = diameter or width of bluff body (Eq. 11-19)
F_a = orifice thermal expansion factor (Eqs. 11-8, 11-9)
F_{am} = correction for air over the water in the water manometer during the differential instrument calibration
F_b = basic orifice factor
F_{gr} = orifice meter specific gravity factor (Eq. 11-6)
F_{hgm} = mercury manometer factor, correction for gas column
F_{hgt} = mercury manometer temperature factor
F_{pb} = pressure base factor (Eq. 11-3)
F_{pv} = orifice supercompressibility factor
F_{pv1} = local gravitational correction for dead weight tester static pressure standard
F_r = orifice Reynolds number factor
F_{tb} = temperature base factor (Eq. 11—4)
F_{tf} = orifice flowing temperature factor (Eq. 11-5)
F_{w1} = local gravitational correction for water column calibration standard
F_{wt} = water density correction (temperature or composition) for water column calibration standard
f = frequency of vortex formation
g = local acceleration due to gravity
H = elevation above sea level (ft)
h_{we} = orifice meter differential pressure (in water)
L = latitude, degrees (0 deg = equator, 90 deg = pole)
MF = turbine meter factor (Eq. 11-16)
P_{atm} = atmospheric pressure (psia)
P = absolute pressure (psia)
ΔP = pressure difference (psi)
Q_h = fluid flow rate (scfh)
Q_{vm} = flow rate through vortex meter (ft^3/s)
R_c = orifice reading on linear chart (Eqs. 11-14,11-15)
R_L = orifice reading on L-10 chart (Eqs. 11-14,11-15)

SG = gas specific gravity relative to air
T = gas temperature (°F)
T_{meas} = temperature (°F) at which orifice plate is measured
t = $D/\cos \theta_f \, v_{son}$ (Eq. 11-25)
t_{AB} = ultrasonic meter downstream transit time
t_{BA} = ultrasonic meter upstream transit time
St = Strouhal number, f db/v (dimensionless)
v = velocity
v_{son} = velocity of sound in fluid (ft/s)
Y = orifice expansion factor
Y_z = expansion compressibility factor
Z = gas compressibility factor

Greek

β = d/D ratio of orifice to pipe diameter
θ_f = $\tan\theta$ = L/D (Fig. 11-17)
μ = fluid viscosity (Ib/ft s)
ρ = fluid density (Ib/ft^3)

Superscript

— = denotes average value

Subscripts

b = denotes base or standard value
f = denotes current case
r = denotes reference or calibration with SG = .6
x = refers to gas with SG = x

MODULE 7:

Process Control & Safety Systems

PROCESS CONTROL AND SAFETY SYSTEMS

PROCESS VARIABLES

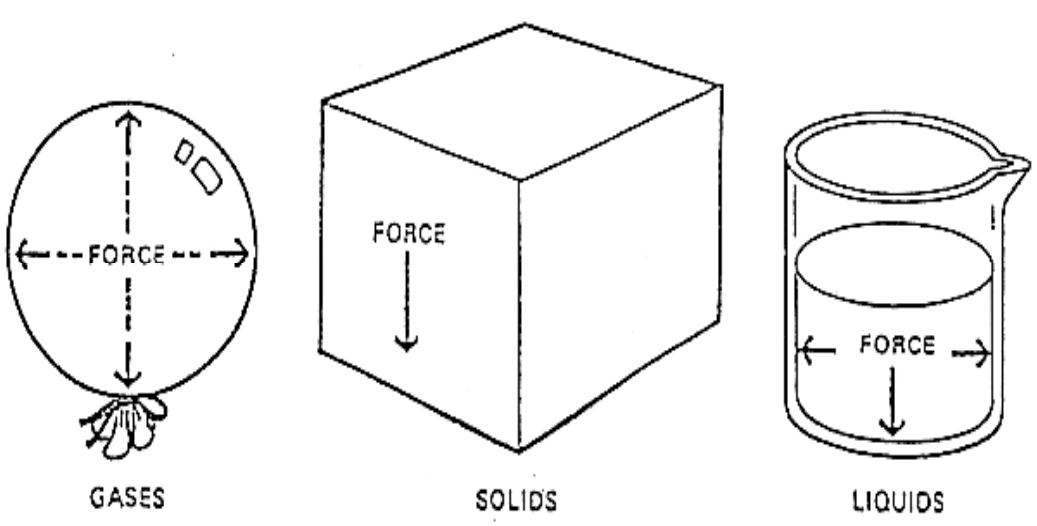
Pressure

Pressure in a process is created by liquids or gases pushing out on the sides of pipelines and vessels, the pressure of a process can affect the properties of hydrocarbons. Boiling points will change under different pressures. Gas-oil separation is affected by pressure. For this reason, pressure must be accurately measured and controlled.

Pressure and Force

Pressure is a measure of force applied to a unit area. Force can be applied in any direction. This makes it different from weight which only applies force in a downward direction.

Figure 1 shows the direction of force exerted by solids, liquids and gases.



The relationship between "Pressure(*P*)", "Force(*F*)", and "Area (*A*)" is expressed as a simple formula:

$$p = \frac{F}{A}$$

Psi, *pounds per square inch,* is the most common-way to describe the unit of pressure.

Atmospheric Pressure

Everything on earth is exposed to atmospheric pressure. We know that air is a gas. We also know that a gas has weight. The atmosphere exerts a force of 14.7 pounds on all surfaces of an object at sea level and at 60^5F. The standard value of atmospheric pressure is 14.7 psi.

Pressure is a very important process variable. Pressure can be helpful. It can also be dangerous. Therefore pressure must be controlled. In order to control pressure, we must first be able to measure it accurately.

Pressure Scales

There are three common scales for measuring pressure:

Gauge Pressure Scale

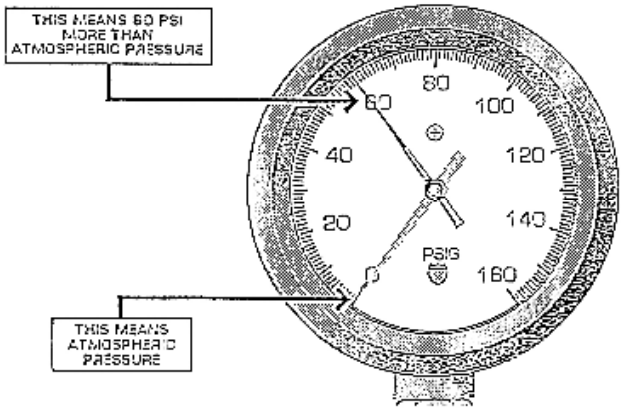
Absolute Pressure Scale

Vacuum Scale

Gauge Pressure Scale

Most pressure measurements in industry are *gauge pressures.* The gauge pressure scale uses atmospheric pressure as the starting point of all measurements. This means that zero on the gauge scale is atmospheric pressure. An unconnected pressure gauge open to the atmosphere will read zero if it is accurate. Gauge pressures are usually measured in pounds per square inch gauge (psig).

Figure 2
Gauge Pressure Scale



Absolute Pressure Scale

The absolute pressure scale uses zero pressure in a perfect vacuum as its starting point. Absolute zero pressure is the complete absence of pressure in a given volume, even atmospheric pressure. This also means the complete absence of matter in a given volume. In practice, it is impossible to have a perfect vacuum.

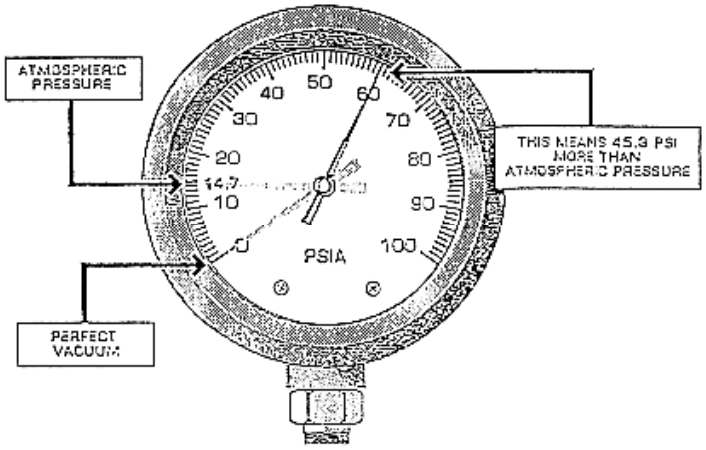


Figure 3
Absolute pressure scale

Pressure readings taken from an absolute pressure scale are expressed as pounds per square inch absolute (psia).

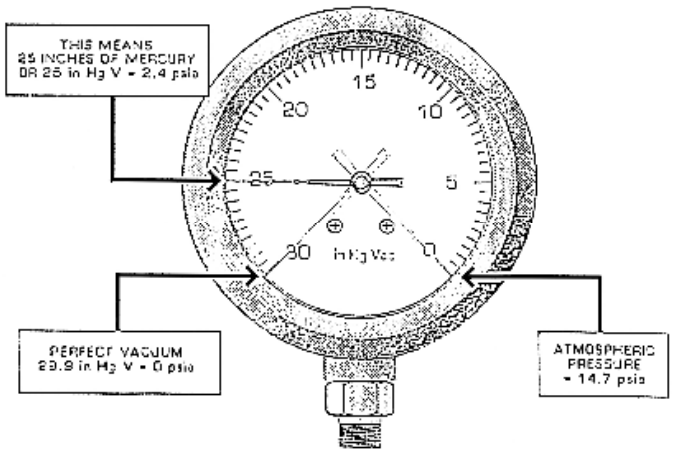
An absolute pressure gauge open to the atmosphere will read 14.7 psia at sea level, if the gauge is accurate. Absolute pressure always equals gauge pressure plus the pressure of the atmosphere. For example, 40 psig will equal 54.7 psia (40 psi + 14.7 psi).

Vacuum Scale

A perfect vacuum is a space that contains absolutely no matter. In hydrocarbon processing plants, vacuum means pressure lower than atmospheric pressure. Vacuum is commonly measured in inches of mercury. A reading of zero on the mercury- vacuum gauge is equal to atmospheric pressure. The maximum value of vacuum in inches of mercury is 29.92 inches. If you had 29.92 inches of mercury vacuum, you would have a perfect vacuum.

One inch of mercury vacuum equals -0.491 psi. In other words, the greater the inches of mercury vacuum, the lower the pressure. For example, 25 inches of mercury vacuum equals a gauge pressure of about -12.3 psig, (-.491 x 25). This equals an absolute pressure of about 2.4 psia. (14.7 - 12.3).

Figure 4
Vacuum Scale



The following chart shows some conversions of vacuum scale readings to gauge and absolute pressures:

IN HG VAC	PSIA	PSIG
0	14.7	0
5	12.2	-2.5
10	9.8	-4.9
15	7.3	-7.4
20	4.9	-9.8
25	2.4	-12.3
29.9	0	-14.7

In industry, there are many ways to express pressure units. Whenever you read a pressure gauge, check it to see what pressure unit the gauge is showing. Pressure gauges will have their measurement units printed on the dial face.

Pressure Measuring Devices

Four types of devices are commonly used to measure or sense pressure:

Manometer

Bourdon Tube Gauge

Diaphragm Gauge

Bellows Gauge

Manometer

A simple mercury manometer is a tube bent into the shape of a U, which contains a measured amount of mercury. When both ends of the tube are open to the atmosphere, the height of the mercury will be the same on both sides. A scale, graduated in inches of mercury, is located behind the tube. The zero mark on the scale is the liquid level line when the pressure at both ends of the tube is the same. .The manometer can be used to compare a pressure with atmospheric pressure or to compare any two different-pressures.

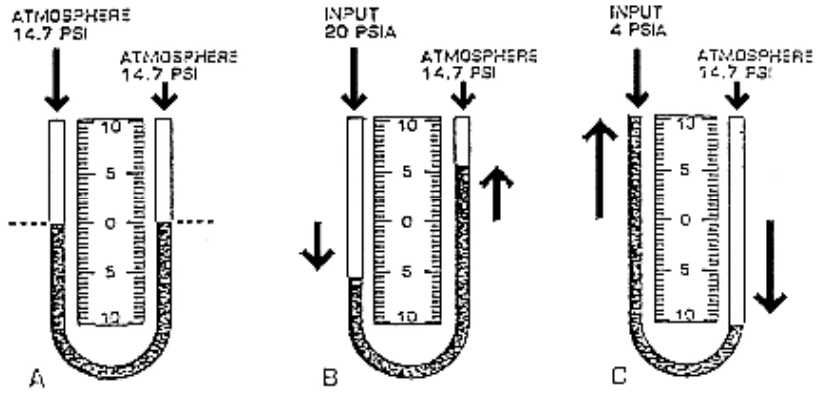


Figure 5

In drawing A, both ends of the manometer are open to the atmosphere. The height of the liquid in both legs is the same.

In drawing B, one leg of the manometer receives an input pressure of 20 psia. Since the input pressure is greater than the atmospheric pressure, it raises the liquid level in the leg open to the atmosphere.

In drawing C, the input pressure is less than the atmospheric pressure. Therefore, the atmospheric pressure raises the liquid level in the leg receiving only 4 psia. ;
The mercury manometer can also measure the difference between two working pressures. The first process pressure is connected to one leg of the manometer tube.

The second process pressure is connected to the other leg. The difference in the height of the mercury in the two legs indicates the *differential pressure* measurement.

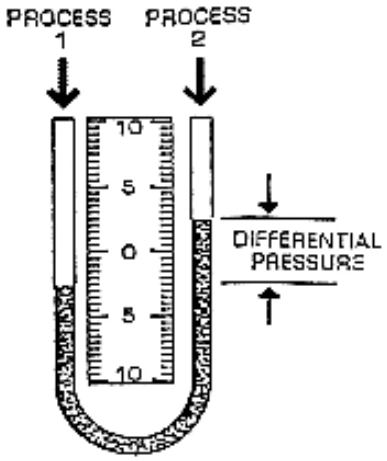


Figure 6

In Figure 6, Process 1 measurement is 2.5 inches of mercury below the zero-mark. Process 2-measurement is 2.5 inches of mercury above the zero mark. When we add the two readings together, the result is a differential pressure of 5 inches of mercury. To convert this to psi, multiply the inches of mercury by 0.491: (5 x 0.491 = 2.46 psi).

Bourdon Tube

The Bourdon tube is a pressure sensing element which changes its shape when pressure is put into the tube.

The Bourdon tube is sealed at one end and connected to a process at the other end. Figure 7 shows how the Bourdon tube responds to pressure. Notice how the tube is partially straightened when .the pressure increases.
Figure 8 shows the Bourdon tube as a part of a pressure gauge. The tip of the tube is connected to a gear mechanism, lever, and pointer.

When pressure is applied to the gauge, the Bourdon tube straightens a little. This movement causes the gear mechanism to change the position of the pointer on the scale. The pointer indicates the pressure measurement.

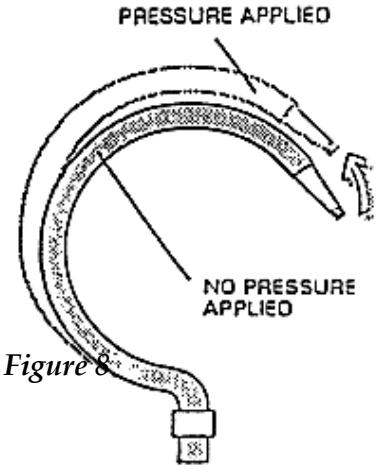
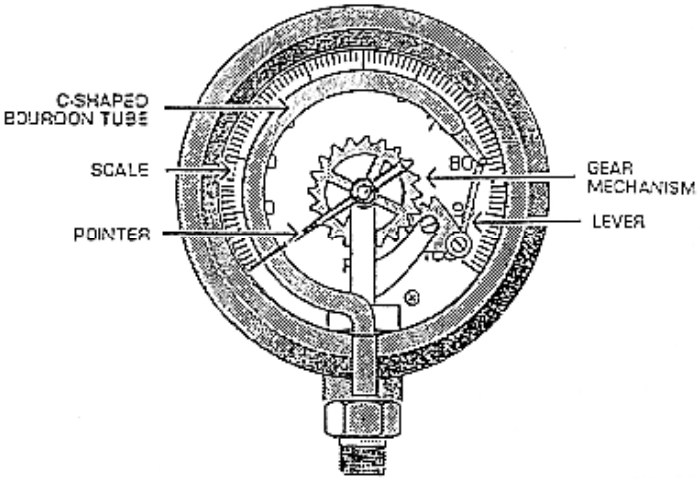


Figure 6

Figure 7



Diaphragm Gauge

This pressure sensing device uses a flexible diaphragm as its sensing element.

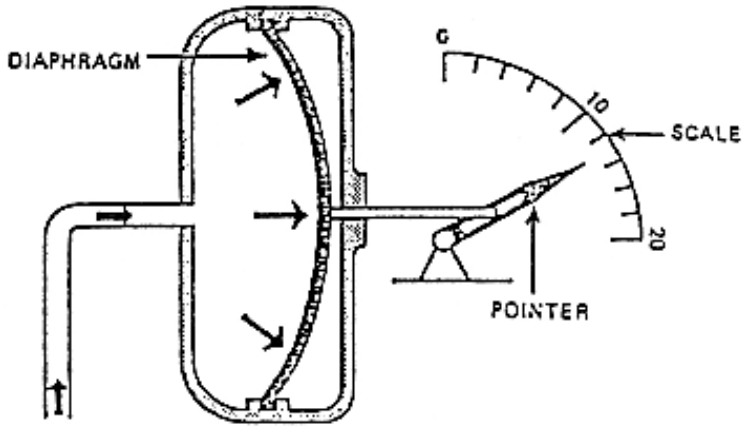
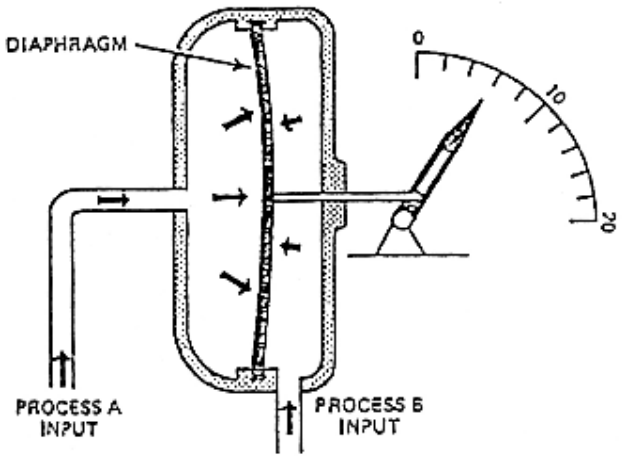


Figure 9

When the diaphragm senses pressure, it moves.-This movement moves a pointer on a scale. The diaphragm gauge is more sensitive than a Bourdon tube.

Diaphragm gauges can also be used to measure differential pressure. The gauge in Figure 10 is designed to compare two process pressures. Input pressures A and B act on opposite sides of the diaphragm. The differential pressure is indicated by the pointer on the scale.

Figure 10



Bellows Gauge

A bellows gauge is sensitive to very small pressure changes.

The expandable element in a bellows gauge is a flexible bellows. A bellows is a small flexible metal can. One end is sealed. The other end is connected to a process.

An increase in pressure causes the bellows to expand. The expanding bellows will move the pointer to indicate a higher pressure.

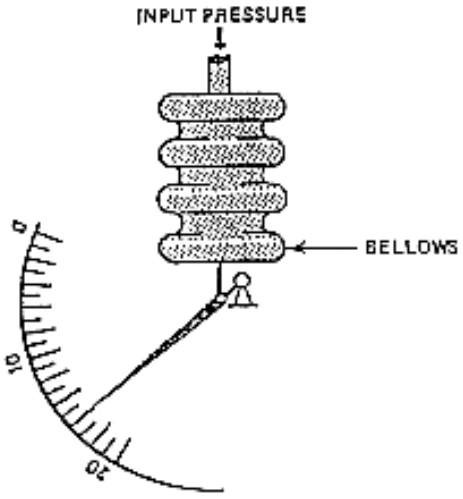


Figure 11

Temperature

Temperature is another important variable in hydrocarbon processing. Fractionation of natural gas and crude oil, for example, requires precise measurement and control of product temperatures.

Temperature is the degree of hotness or coldness, measured on a definite scale.

The motion of the molecules in a substance determines how hot or cold it is. When the molecules move rapidly, they strike against each other more often. This causes a release of heat, raising the temperature. When the motion of the molecules slows, the temperature falls.

Temperature must be closely monitored because it is difficult to control. This is because it is slow to react. When something is too hot, it takes time to cool. When it is too cold, it takes time to heat up.

Heat Transfer

When two substances of different temperature come into contact with each other, there is a flow of heat. This flow is called heat transfer. Heat transfer will always be *from* the hotter substance to the colder substance.

Heat is transferred in three ways:

Convection

Conduction

Radiation

Convection

Convection is the transfer of heat by moving molecules in gases or liquids by *natural circulation.*

Figure 12 shows an example of heat transfer by convection. As hot air molecules flow from the heater vent, they rise. As they cool, they drop and return to the furnace, where they are heated and then circulated again.

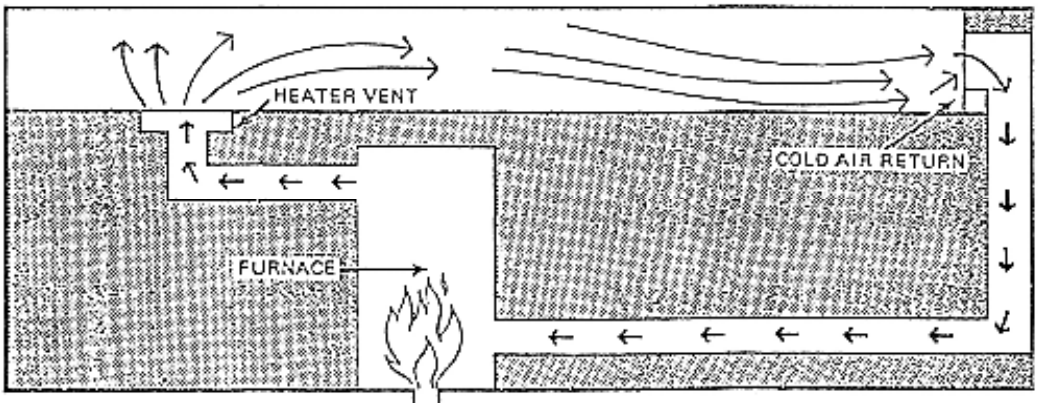


Figure 12

Conduction

Conduction is the transfer of heat *through* or *across a conductor.*

Heat is transferred by the direct contact of the molecules in the conductor. Figure 13 shows an example of conduction. Here, the heat is transferred along the steel rod. Conduction explains why the metal handle of a pot gets hot when it is not directly exposed to heat.

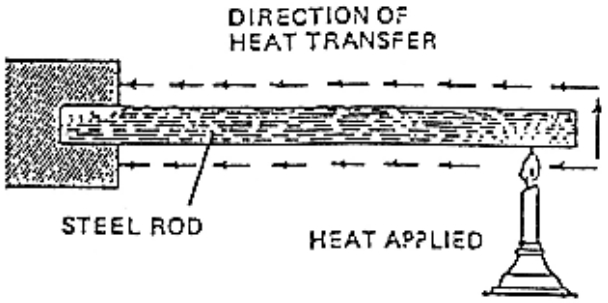


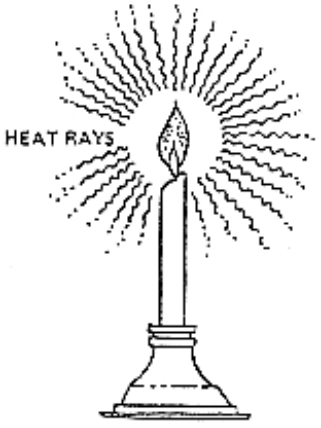
Figure 13

Radiation

Radiation is the transfer of heat by *rays.*

The radiating source gives off heat waves which are absorbed by a colder substance or object. This is how the sun heats the earth's atmosphere. It explains why objects left out in the sun get too hot to touch. Figure 14 shows an example of heat transfer by radiation.

Figure 14



Temperature Scales

Two scales are commonly used to measure temperature. They are the *Fahrenheit* scale and the centigrade or *Celsius* scale.

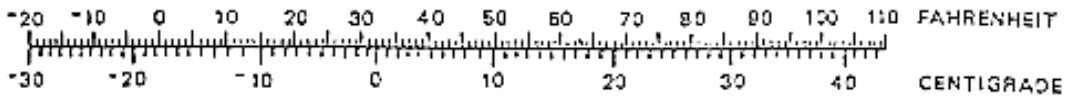


Figure 15

The Fahrenheit (°F) and Celsius (°C) scales are based on the freezing and boiling points of water. On the Fahrenheit scale, water freezes at 32° and boils at 212° under atmospheric pressure. On the Celsius scale, water freezes at 0° and boils at 100° under atmospheric pressure.

Temperature Measuring Devices

Five types of devices are commonly used to measure or sense temperature:

Mercury Thermometer

Filled-System Temperature Indicator

Bimetallic Thermometer

Thermocouple

Resistance Temperature Detector

Mercury Thermometer

A mercury thermometer consists of a bulb filled with mercury attached to a capillary tube with all the air removed. A capillary tube is a very thin tube. When the temperature rises, the mercury in the bulb *expands*. The expansion causes the mercury to rise in the capillary tube.

A column of mercury in a glass tube does not form a flat surface at the top of the column. The mercury bulges up in the center, as shown in Figure 16.

To read a mercury thermometer (or a mercury manometer), you must use the highest point of the mercury in the column. In Figure 19, the correct reading is 47° F.

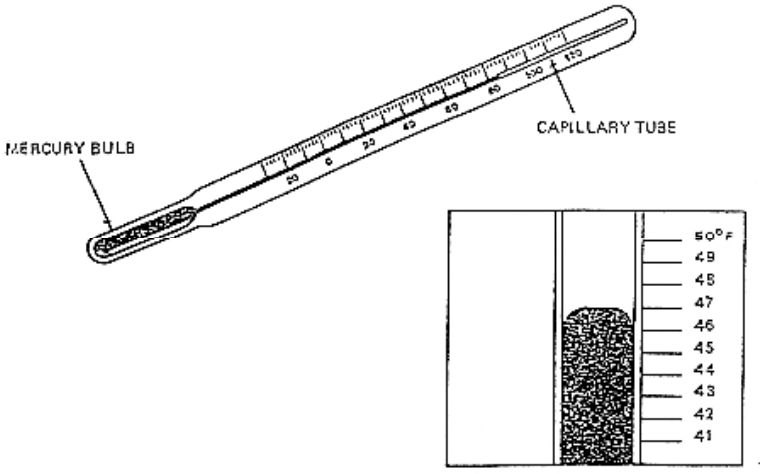


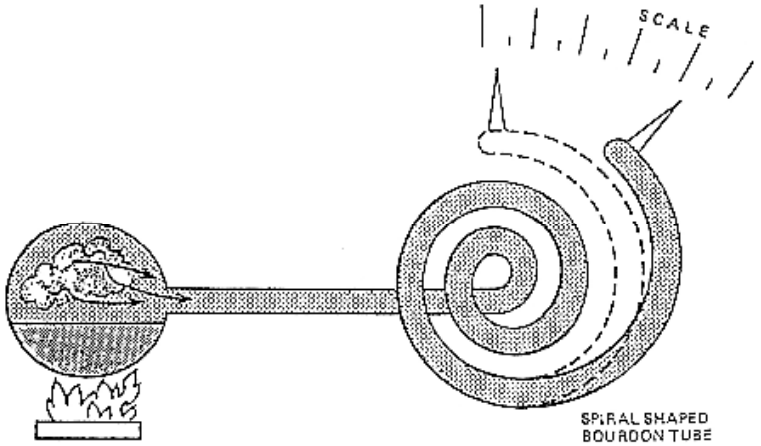
Figure 16

Filled-System Temperature Indicator

When the temperature in a closed container increases, the pressure in the container increases. The pressure changes in the container can be measured to indicate the change in temperature. This is the principle of a filled-system temperature indicator. A scale on the indicator is marked to convert pressure changes to temperature changes.

The bulb in a filled-system temperature indicator can be filled with liquid and vapor. When the temperature rises, more liquid is changed to a vapor. The increased vapor pressure straightens the spiral Bourdon tube pressure sensor. The movement of the Bourdon tube, caused by a pressure change, is read as a temperature change.

Figure 17



Bimetallic Thermometer

Metals expand (increase in size) when heated, and contract (decrease in size) when cooled. Different metals expand at different rates. This property is used in bimetallic strips.

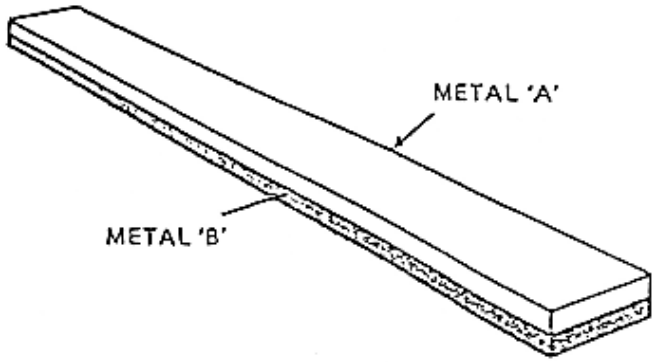


Figure 18

A bimetallic temperature device operates on the principle of *thermal expansion.* A bimetallic device contains two different metallic alloys fused together. when the two metals are heated, they expand at different rates. This causes the strip to bend as shown in Figure 19.

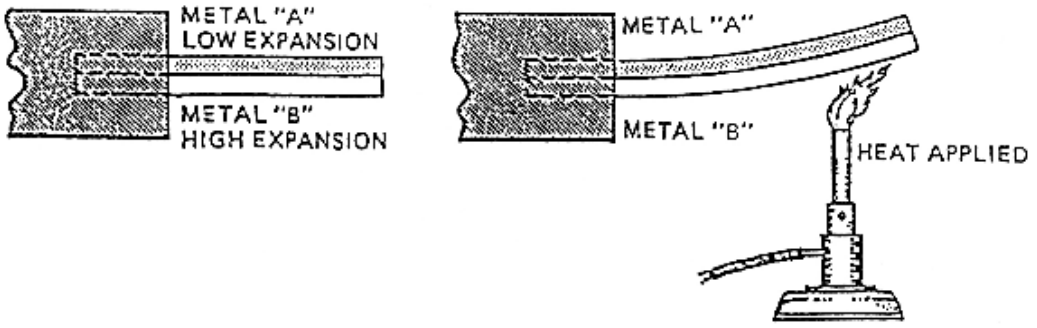
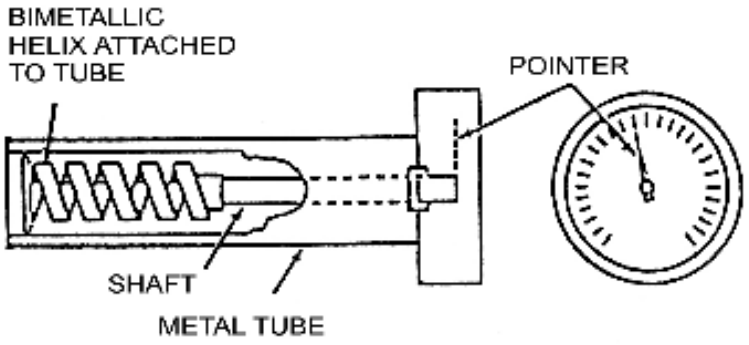


Figure 19

Figure 20 shows a temperature measuring device which uses a helical (coil) bimetallic element.

When the coil is heated, it unwinds because of the different expansion properties of the two metals. A shaft connects the helix to a pointer. As the helix unwinds, it moves the pointer to indicate the temperature.

Figure 20



Thermocouple

A thermocouple is a temperature sensing device that generates an electrical voltage when it is heated. It consists of two wires made from different metals.

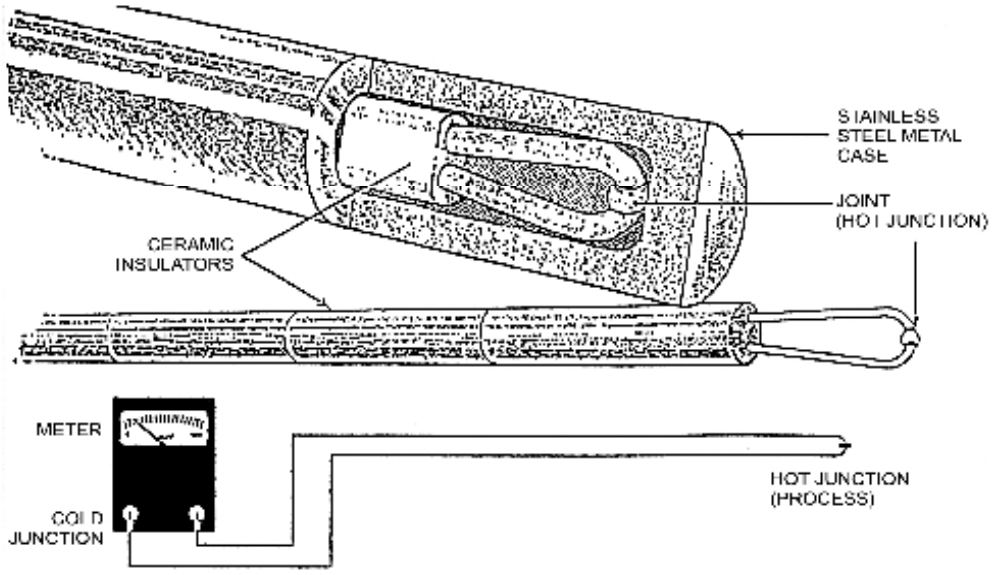


Figure 21

The wires are joined at the measuring end (hot junction) of the thermocouple. At the other end, the wires are connected to a meter that senses very small amounts of voltage. Even if the temperature of the measuring end is changed, the end connected to the meter remains at a constant temperature. The difference in temperature between the hot and cold junctions produces a voltage that can be read on the meter as a temperature.

Resistance Temperature Detector

Resistance is opposition of a material to the flow of electrical current. Changes in temperature will affect the electrical resistance of certain materials. A resistance temperature detector (RTD) makes use of this property of materials.

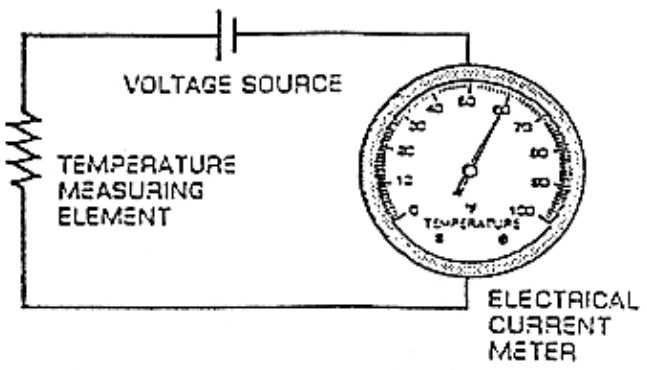


Figure 22

An electric current flows through a temperature measuring element. As the temperature measuring element heats up, its resistance to current flow increases. .
The electric current flow drops as-the resistance of the element increases. A meter can be used to detect the change in current flow caused by changes in temperature. The meter reading will correspond to the Temperature of the measuring element.

Flow

Flow is simply the movement of a liquid or gas from one place to another. It could be gas flowing through a pipeline or crude oil flowing up from under the ground. The *rate of flow* tells an operator the amount of product moving through a plant or system. Flow is an important variable in process operations. Most equipment is designed to operate within a minimum and maximum rate of flow. Rate of flow can also affect temperatures and pressures in a process operation.

Flow Rate

Flow is measured by the amount of liquid or gas that moves past a point in a certain time. This is called *flow rate.* Flow rate is usually expressed in gallons per minute, barrels per hour, or thousands of barrels per day for liquids. Flow rate for gases is usually expressed in standard cubic feet per hour.
Two factors control flow rate:

The size of the pipe the fluid flows through.

The amount of pressure difference between the ends of the pipe.
Liquids and gases are *fluids.* When the word fluid is used, it can refer to liquids or gases, or both.

Flow Measuring Devices

Flowmeters are installed in pipelines to measure the amount of fluid passing through the line.

Four basic types of flowmeters are used in plants:

Velocity Flowmeter

Positive Displacement Flowmeter

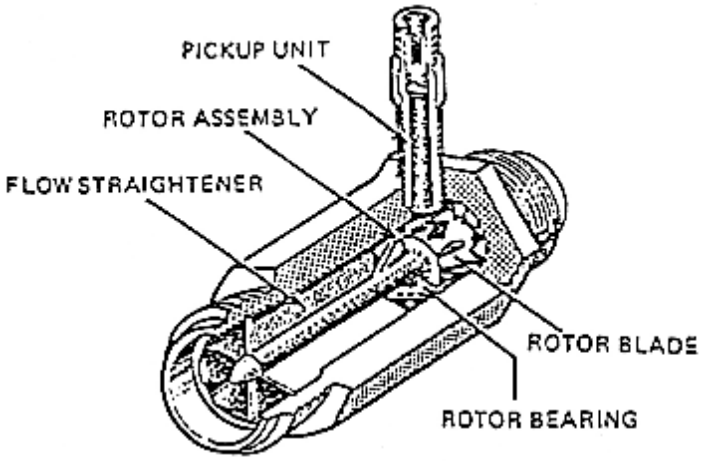
Differential Pressure Flowmeter

Variable Area Flowmeter

Velocity Flowmeter

A velocity flowmeter is a rotating device. It allows a specific amount of fluid to pass for each revolution of the rotor. The number of revolutions in a given time indicates the flow rate. The higher the flow rate, the faster the rotor turns.
The turbine flowmeter is the most commonly used velocity flowmeter. The turbine flow-meter is subject to high wear. It must be used with clean fluids only.

Figure 23



Positive Displacement Flowmeter

An impeller meter is a positive displacement flow-meter that uses two rotating impellers. Each time the impellers make a complete revolution, a definite amount of fluid is allowed to pass. The number of revolutions of the impellers in a given period of time is counted automatically to find the flow rate. The higher the flow rate, the faster the impellers turn.

The impeller meter has close fitting parts. This type of meter can be used only with clean fluids. Fluids which contain abrasive particles, like sand cause excessive wear. Wear causes the meter to be inaccurate.

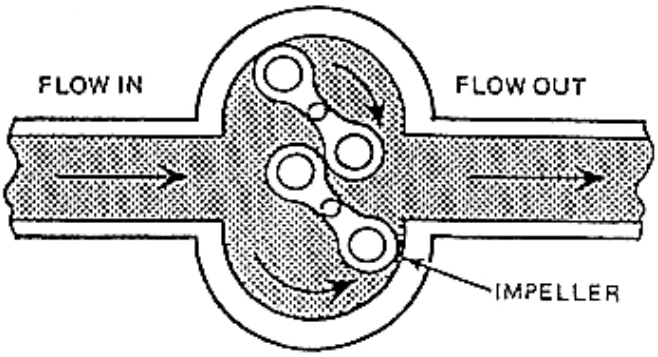


Figure 24

Differentia Pressure Flowmeter

Forcing a liquid or gas through a reduced area (or *restriction)* creates a localized area of high pressure on the upstream side, and low pressure on the downstream side of the restriction. As the fluid continues through the pipeline, the pressure recovers almost fully.

The higher pressure upstream and the lower pressure downstream of the restriction create a differential pressure in the pipeline. The greater the flow rate, the higher the differential pressure. The differential pressure value is measured and converted to a flow rate. Figure 25 illustrates the differential pressure principle.

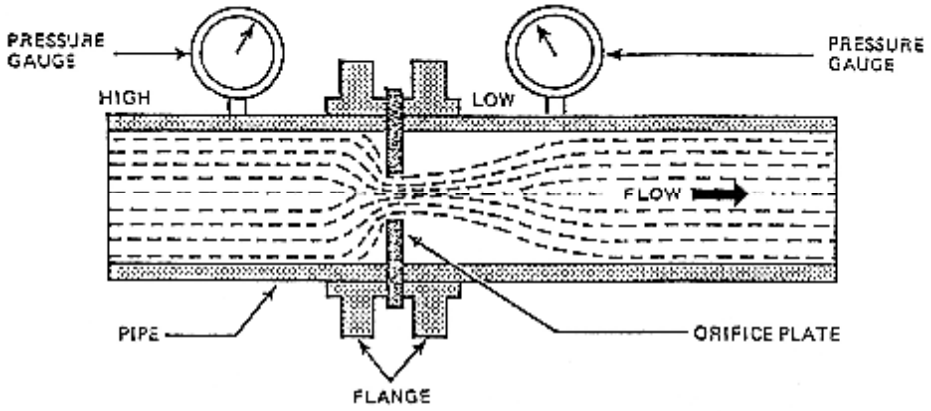


Figure 25

Three common devices are used to create differential pressure in pipelines. They are the:

Orifice Plate

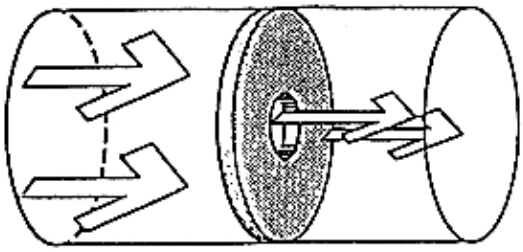
Flow Nozzle

Venturi Tube

Orifice Plate

An orifice plate is a thin metal plate with a hole in it. The plate is put into a pipeline to create a restriction. An orifice plate is the most common device used in measuring the flow rate of a liquid or gas through a pipeline.

Figure 26



Flow Nozzle

The flow nozzle is another restricting device used for differential pressure measurement.

The flow nozzle works in a similar way to the orifice plate. It creates a localized area of high pressure in the pipeline upstream of the nozzle, and an area of low pressure downstream of the nozzle. The differential pressure is the difference between the upstream pressure and the downstream pressure.

A flow nozzle is more expensive and more difficult to install than an orifice plate. The flow nozzle must only be used with clean liquids.

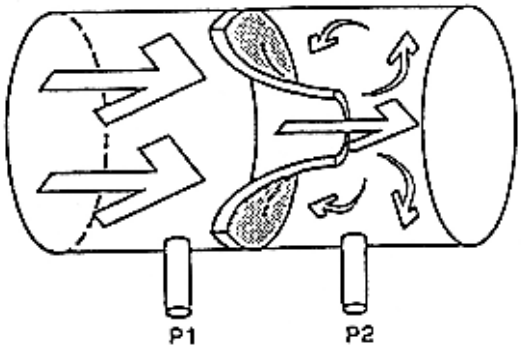


Figure 27

Venturi Tube

Another pipeline restriction used for flow metering is the Venturi tube. This type of device is more accurate than the orifice plate and the flow nozzle The **Venturi** tube is expensive and more difficult to install in a pipeline than the orifice plate or the flow nozzle. The Venturi tube has the advantage that it can be used with dirty liquids. Solid particles that cause wear in orifice plates and flow nozzles pass through a Venturi tube causing little or no wear

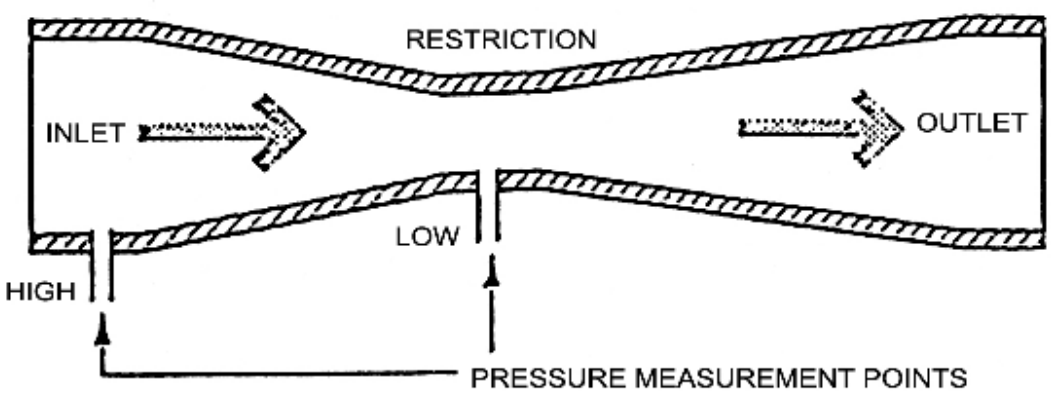
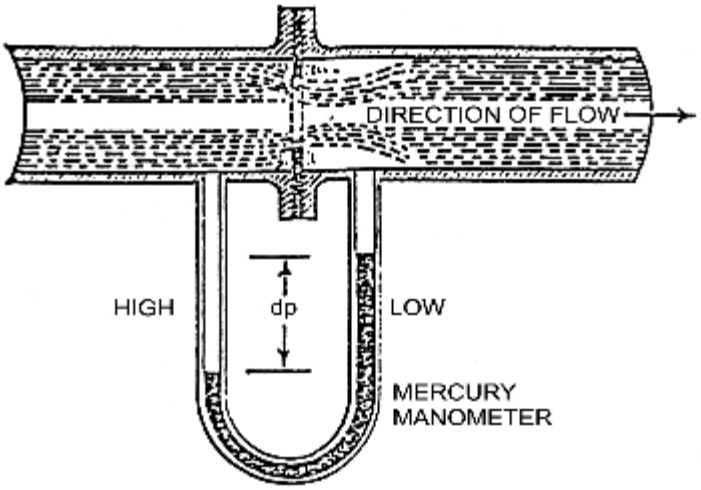


Figure 28

The orifice plate, the flow nozzle, and the Venturi tube are restrictions which create a differential pressure. We need a device to measure the differential pressure. Figure 29 shows a manometer set up to measure differential pressure across a flow restriction.

Figure 29



Variable Area Flowmeter

Variable area flowmeters are used in pipelines where the flow rate is low, such as in chemical treatment systems.

The rotameter is a variable area flowmeter. It works very differently from other differential pressure flow metering devices. The rotameter uses a varying restriction to keep the difference in pressure constant. The varying restriction is made of a tapered tube and a float. Figure 30 shows a rotameter with a low flow rate on the left and a higher flow rate on the right.

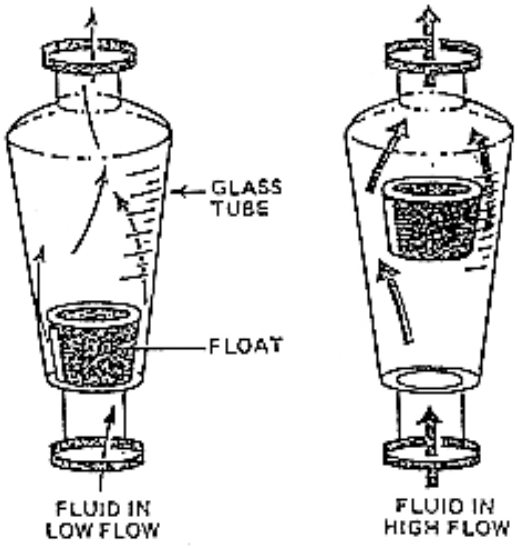


Figure 30

The metering tube is mounted vertically. The narrow end of the tube is at the bottom of the rotameter. Fluid enters the bottom of the tube and flows upward around the float. The fluid leaves-through the wide end of the tube at the top of the rotameter.

The flow rate is indicated by the position of the float in the tube. If there is no flow through the tube, the float remains at the bottom of the tube. As flow begins, the fluid forces the float to rise in the tube. The float will rise until the flow of fluid produces equal pressure at each end of the float. Since the float is made of a material that is denser than the process fluid, it does not actually float. The float stays suspended in the fluid because of the pressure developed by the fluid flowing through the tube.

Level

Level refers to the depth of a liquid in a tank or container. Any tank or vessel in a process plant is designed with a maximum and minimum level. The product must not go above or below these limits. Operators must constantly monitor levels in this equipment.

Level Measurement

The first level indicators were probably sticks. The sticks were placed in streams or pools of water to find out how deep the water was. We still use this method of checking level today. An example of a level stick indicator is the dipstick in an automobile engine. It is used to check the level of the oil.

Level measurement is important in process plants where crude oil, water, chemicals, and other materials move through or are stored in large tanks or containers.
You can see the level in an open container. But many containers are not open. We need a way of checking the level in closed containers.

Level Measuring Devices

Five common types of level measuring devices used in plants are:

Hand Reel Tape

Sight Glass Indicator

Bubbler Level Indicator

Float-type Level Indicator

Displacer Level Indicator

Hand Reel Tape

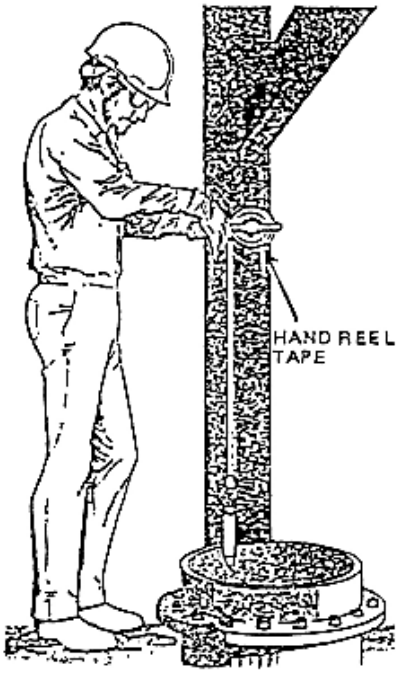
The hand reel tape is the simplest and most accurate way of measuring the level of liquid in a tank.

Hand reel readings must be corrected for thermal expansion of the liquid. A correction table is used that is based on a standard temperature of 60°F
The hand reel tape has a disadvantage. It can be used only in tanks that have an opening at the top of the tank. It cannot be used in sealed tanks that contain highly volatile liquids.

The hand reel tape is a steel tape with a weight on the end. The tape is wound on a metal spool with a hand crank. This device is carried from tank to tank.

When the weight is lowered manually into the tank, most liquids cause a wet mark on the tape. The mark can then be read when the tape is pulled up.

Figure 31



Sight Class Indicator

A sight glass (or gauge glass) is a tube mounted on the outside of a tank. It usually connects to the top and bottom of the tank. The level on the sight glass matches the liquid level in the tank. When the level in the tank rises or falls, so does the level in the sight glass.

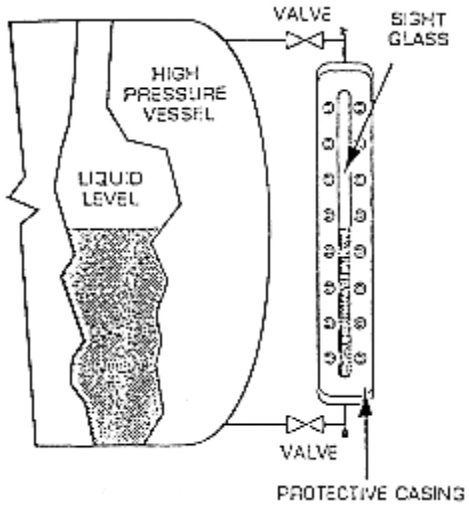


Figure 32

Bubbler Level Indicator

Bubbler level indicators are used in tanks that contain corrosive liquids, such as chemicals.

In a bubbler system, a tube is placed in a tank containing liquid. Air is forced down the dip tube so it bubbles out of the bottom. The deeper the liquid in the tank, the more pressure it takes to push the air out of the bottom of the tube. This pressure value can be converted into a level reading.

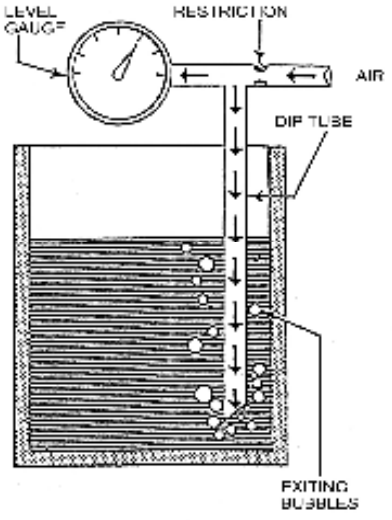


Figure 33

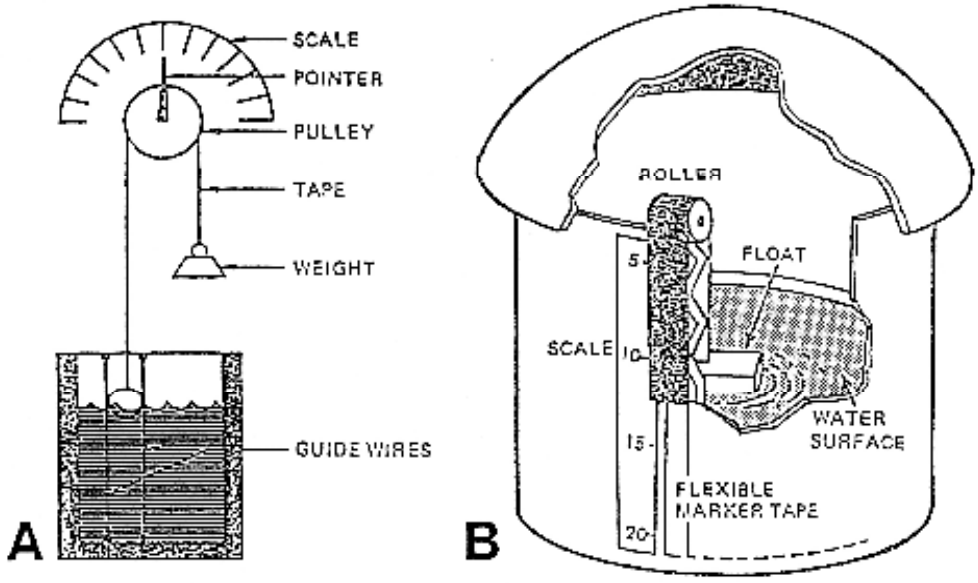


Figure **34**

Figure 34 shows two ways that the movement of the tape or wire is used to show the liquid level in a tank.

In Figure 34A, the wire moves a pulley. A pointer is attached to the pulley. The up or down movement of the wire attached to the float turns the pulley. As the pulley turns, the pointer indicates the level in the tank on a measurement scale.

In Figure 34B, one end of the tape is connected to a float in the tank, and the other end is connected to a weight. The weight keeps the tape tight. As the level in the tank changes, the weighted end of the tape moves up and down on the graduated scale. The end of the tape shows the liquid level.

All of the level indicators we have discussed so far are used in storage tanks. Now we will look at a liquid level measurement device that is commonly used in process vessels.

Displacer Level Indicator

[

A displaper level system is used to measure a small range of liquid level, such as in a process vessel. The system operates on the principle of buoyancy-. This is the power of a fluid to exert an upward force on a body placed in it.

A liquid exerts an upward push on an object placed in it. This makes the weight of the object appear to be less. The weight of the object is reduced by the weight of the liquid it displaces.

A ship floats because it displaces as much water as the ship weighs. Buoyancy explains why a tanker which weighs 100,000 tons will float instead of sinking to the bottom like a stone. The tanker displaces 100,000 tons of water.

In Figure 35A, you can see that the displacer weighs 3 pounds. In Figure 35B, 1 pound of liquid has been displaced. The displacer now seems to weigh 2 pounds.

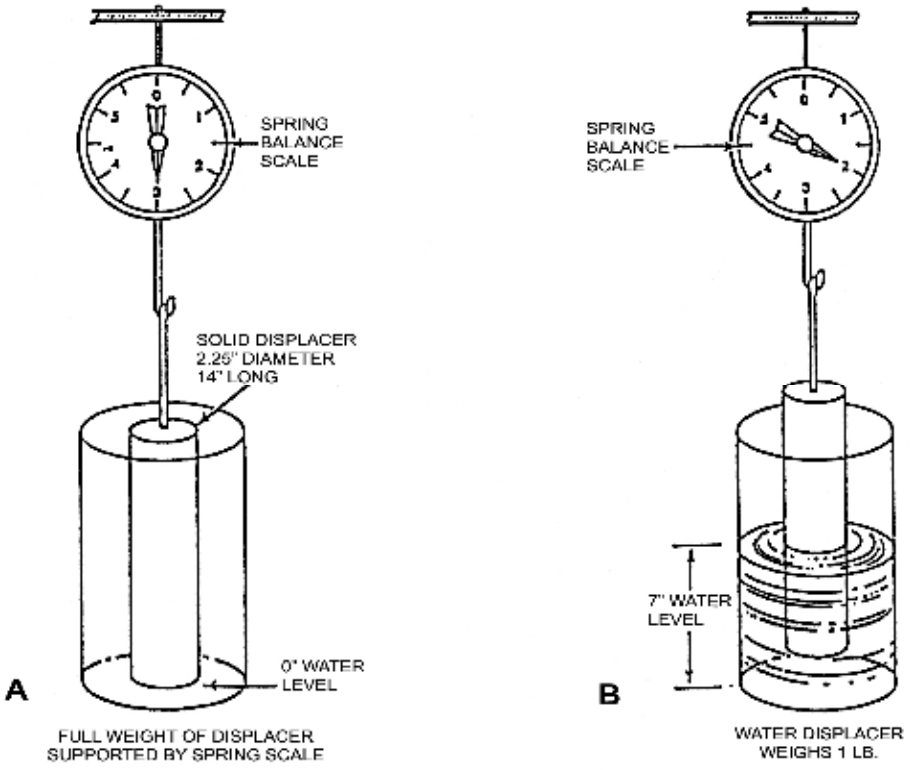


Figure 35

Figure 36 shows how the buoyancy principle is put to work in a displacer level indicator.

When the liquid level is low, the weight of the displacer twists or torques the shaft. This moves the pointer to show the low level on the indicator scale.

When the liquid level rises, buoyancy makes the displacer weigh less. The shaft has less and less twist as the liquid level increases. This moves the pointer to show a higher liquid level.

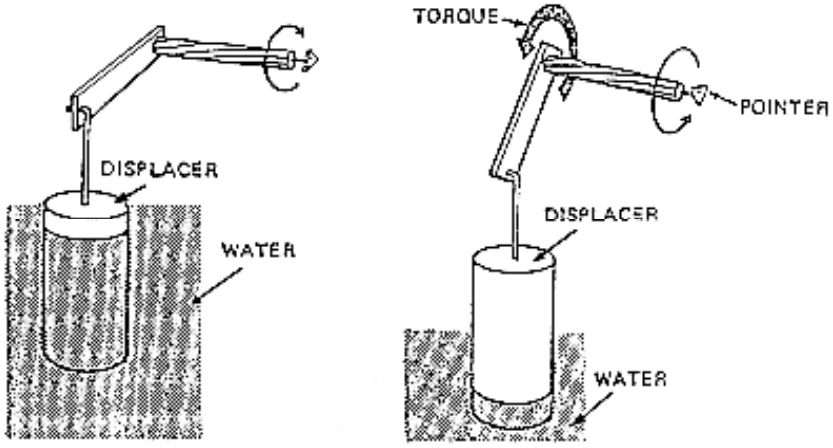


Figure 36

Very sensitive electronic installments can be connected to a displacer measurement system. These instruments can then be used to control the level in a process vessel.

PROCESS CONTROL

In early days of industrial development, the control of processes was carried out manually by operators. This operation often depended upon the senses of sight, smell, sound and touch which were developed with experience by the personnel involved. For many years process operators spent a large proportion of their working period in adjusting valves during tours of the plant. These tours would be repeated at regular intervals and appropriate notes recorded on a log sheet. The quality of control resulting from this procedure was comparatively poor by present day standards as it relied on developed skill and acquired knowledge.

In more recent times, the growing complexity and multiplicity of modern plants necessitated a form of control which could overcome the human limitations. To meet this need, automatic control systems were developed. Some of the early systems consisted of relatively simple self-acting controllers. More complex pneumatic and electronic systems are now in common use, and control may be achieved by means of computers. Using techniques involving, for example, mathematical models, it is possible to predict the behavior of a process with a high degree of accuracy. From this information automatic controllers can be adjusted to give optimum performance, replacing earlier procedures by a highly scientific approach.

Control Loops

A number of operations, one dependent upon another, ore involved in most control systems and these are often represented for convenience by a block diagram. Such a diagram shows the control loop for the installation concerned. It should be fully appreciated that the process itself forms a significant part of the loop.

Open loop system

A typical system is arranged for the value of the controlled condition to be continuously shown on an indicating or recording instrument. A reading of the instrument is taken periodically and the valve V adjusted accordingly. Thus the output of the measuring element does not play a direct part in actuating the valve, its only function being to provide information. As this arrangement does not form a complete loop, it is referred to as an open loop system and its practical applications are limited.

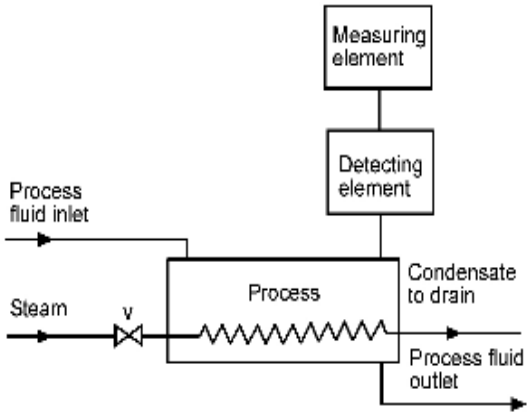


Figure 37

Closed loop systems

In a closed loop system, a complete loop is formed which starts at the process then via the detecting element, controller and valve back to the process. In contrast to the open loop system, the measured value is fed back to the controller and, because of this, the term feedback control is used to describe ,the closed loop system. The controller continuously adjusts the valve to maintain the controlled condition at the desired value

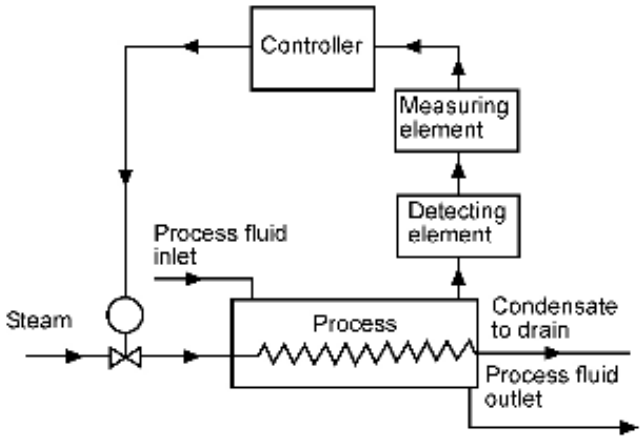


Figure 38

Automatic control

Due to the growing multiplicity and complexity of modern manufacturing processes, it was realized that some form of automatic control would be desirable.

The advantages of automatic control include:

The process operators are relieved of the more onerous tasks thus making better use of manpower.

Steady conditions reduce the waste of raw materials and fuel.

Plant can be operated nearer to its maximum capacity than it can be with manual control.

The superior control improves the quality of the product.

A more uniform product is obtained and thus it can conform closer to a given specification.

Processes can be operated which could not be manually controlled.

Most automatic control installations are closed loop systems and components of a typical loop are shown schematically.

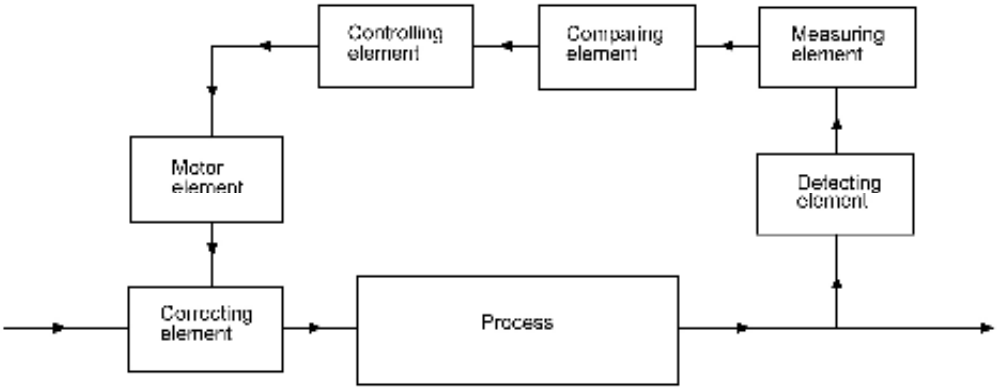


Figure 39

Basic Control Systems

Process variables can be automatically controlled in a number of different ways. Details of the system to be used are decided after consideration of vital factors, eg, the conditions which will be established should a failure occur. The system selected will be one which offers a maximum degree of safety to personnel and plant.

Pressure Control

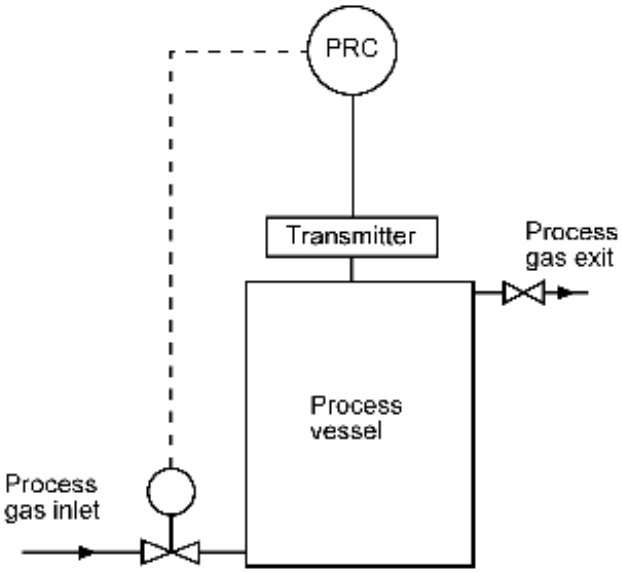
In a typical pressure control system, the correcting element is connected in the inlet line to the process vessel. The load, ie, the throughput of gas, is varied by manipulation of the hand-operated valve in the exit line. A signal related to the vessel pressure is transmitted to the pressure recording controller (PRC) and the controller, in turn, generates an output signal which is used to actuate the correcting element. Thus, the amount of gas admitted is just sufficient to maintain the pressure in the vessel at the desired value.

As an alternative, control can be achieved by installing the correcting element in the exit line. In this system, more gas is released from the vessel when the pressure rises above the desired value.

In some vacuum systems, air from the atmosphere is admitted through the correcting element in order to control the absolute pressure. The nature of the process fluids determines where this technique can be applied.

It is usually desirable to have the suction pressure of a compressor maintained at a constant value. This is accomplished by using the correcting element either as a restriction to the flow through the compressor, or as a by-pass between the delivery and suction sides.

Figure 40

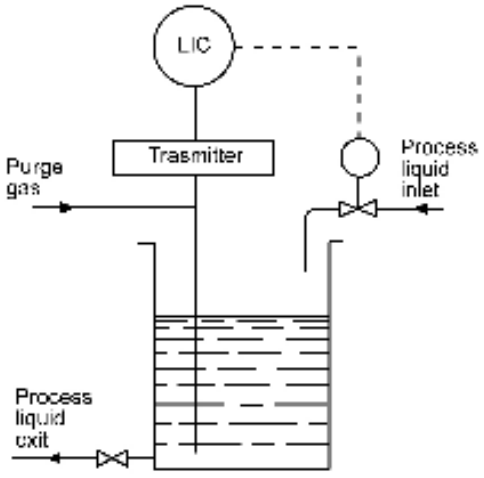


Level control

In a typical level control system, the transmitter measures the pressure established in the purged dip pipe, this pressure being proportional to the liquid level. Load is changed by means of the hand operated valve located in the tank exit pipeline. The output signal of the indicating level controller (LIC) actuates the correcting element, and the amount of liquid which is allowed to flow into the tank maintains the level at the desired value.

To form another system the correcting element may he installed in the tank exit pipeline. In this instance, the amount of liquid flowing from the tank is controlled to maintain a constant level.

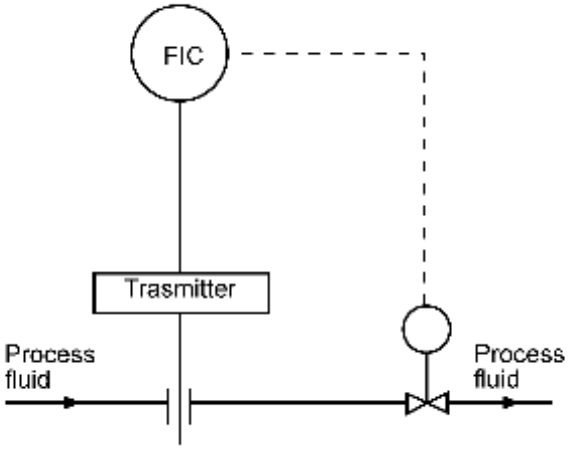
Figure 41



Flow control

In a typical flow control system, the transmitter measures the differential pressure developed across the primary element, this differential pressure being related to the flow. It is important that the primary element, eg, an orifice plate, is installed in a region where changes in pressure are minimal. This requirement applies particularly to gas flow applications since the measurement is correct at one specified value of pressure. The most suitable position is, in general, upstream of the correcting element. In operation, adjustment of the flow through the correcting element continues until it coincides with the desired value.

Figure 42



Temperature Control

Temperature control can be effected by either supplying or removing heat. In a typical system, the control depends upon variations being made to the amount of heating medium supplied. When the temperature rises above the desired value, the correcting element reduces the heat input accordingly.

Reaction processes which generate heat are examples of systems requiring control by the alternative method. For effective temperature control of these processes, the excess heat must be removed by a controlled supply of a cooling medium such as water.

Note: Pressure, level, flow and temperature controllers are in common use. The control of other process variables such as density, viscosity, pH, Eh, gas composition and liquid conductivity is also carried out.

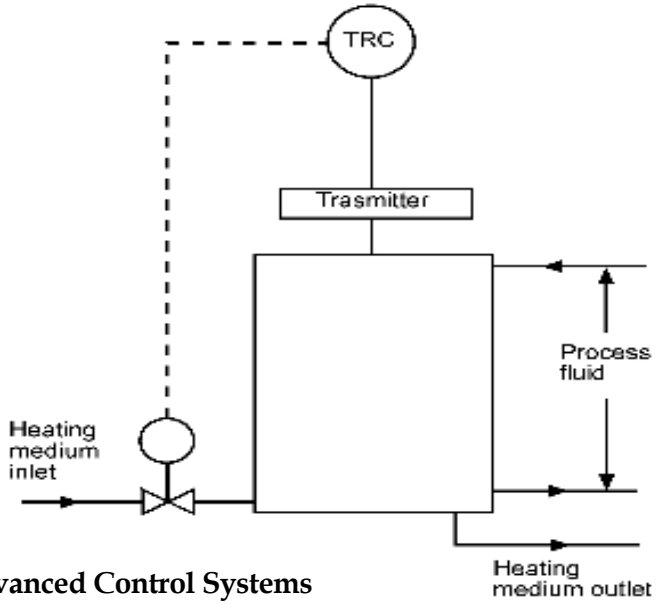


Figure 43

Advanced Control Systems

Cascade Control

In a cascade control system applied to a heat exchanger, variations in the flow rate of the heating medium, due, for exam pie, to changes in steam pressure, is a source of disturbance. Hi is flow rate is con trolled by the secondary controller FRC and so the disturbance does not enter the system. Variations in temperature are corrected by the action of the primary controller TRC. The output of this controller is used to adjust the set point of the secondary controller. This causes a change in the signal to the control valve and so the required corrective action is achieved. When cascade techniques are employed on this type of system, stable control is obtained.

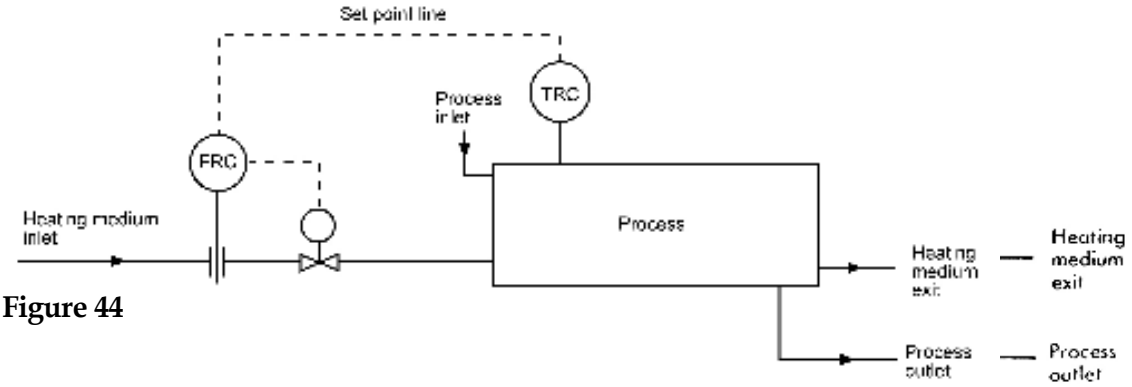


Figure 44

Ratio Control

It is sometimes necessary to control the ratio of two process fluid flow rates and, in one method by which this may be accomplished, the flow of fluid B is controlled by FRC and a transmitter is installed to measure the flow rate of fluid A. Assuming that the system is in equilibrium and then the flow rate of fluid A alters, the result is a change in the transmitter output signal. This output signal is used to adjust the set point of the flow controller and, in turn, the controller output signal actuates the valve.

Adjustment of the valve continues, due to the design of the controller mechanism, until the flow of fluid B reaches a value which conforms to the original ratio. The ratio is varied by means of an adjuster, provided in the controller mechanism, which alters the relationship between the input and the output. For some flow ratio systems, it is essential that the transmitter output shall have a linear relationship to the measured flow and this requires the inclusion of a square root extractor.

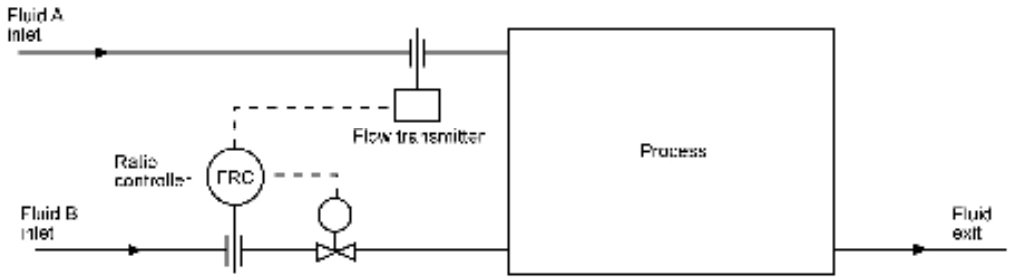
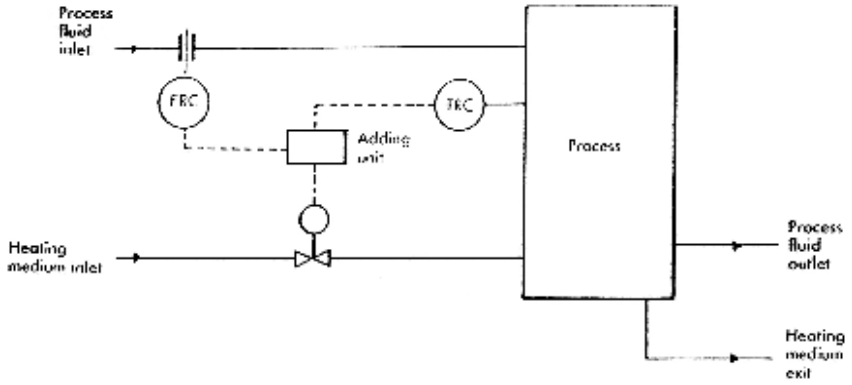


Figure 45

Feed forward control

The effect of disturbances on the controlled condition can sometimes be accurately ascertained. For example, the changes in the temperature of a system caused by variations in process fluid flow may be determined. By feeding forward this information, the effect of disturbances can be reduced. An example of feed forward technique is illustrated.

Figure 46



In this instance, the purpose of the system is to control the temperature at a point in the process vessel, because variations in process fluid flow rate are a major source of disturbance. The output signal of controller FRC changes when the process fluid flow rate alters. This output signal is fed forward to an adding unit where it is added to the signal generated by feedback controller TRC. Corrective action thus takes place before the temperature is affected. The controller FRC must not contain integral action, this limitation being necessary in order to avoid the possibility of a small sustained deviation causing integral saturation.

Control Valves

In process systems, the final control element is normally a pneumatically actuated control valve, which is used to regulate the flow of a fluid. It provides the necessary power to translate the controller's output to the process. Pneumatics are used because of the original popularity of pneumatic control systems and the comparatively low operating pressures used.

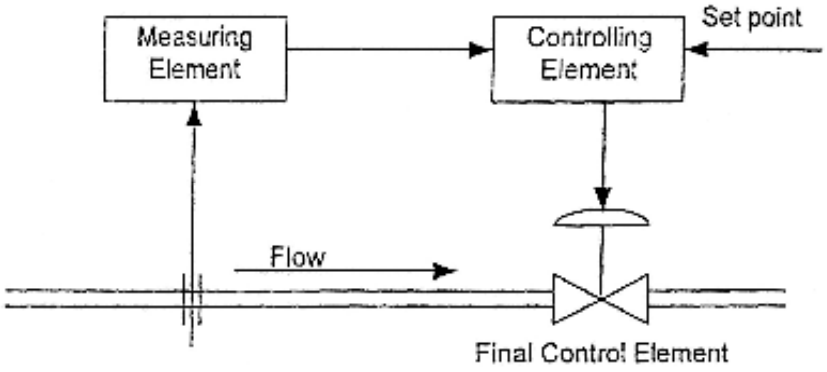


Figure 47

Of the basic components of a control loop the control valve is subject to the harshest conditions. A control valve is also the most expensive item and the most prone to incorrect selection.

Control Valve Selection

The choice of control valve will depend upon the application. i.e. flow control, ESD etc.

The main factors to take into account are the valves' ability to regulate flow, the pressure loss/recovery when fully open and the shut-off leakage. For instance a globe valve gives good flow regulation, has poor pressure recovery at high flow rates and does not give tight shut-off, whereas a ball valve has poor flow regulation

characteristics, low pressure loss at high flow rates and has the advantage of tight shut-off. Space is another factor that can come into the equation.

The main selection factors of control valve type can be summarized as follows:

The process requirements:
1. inlet pressure
2. Differential pressure across the value
3. Maximum and minimum flow rates
4. Degree of shutoff .

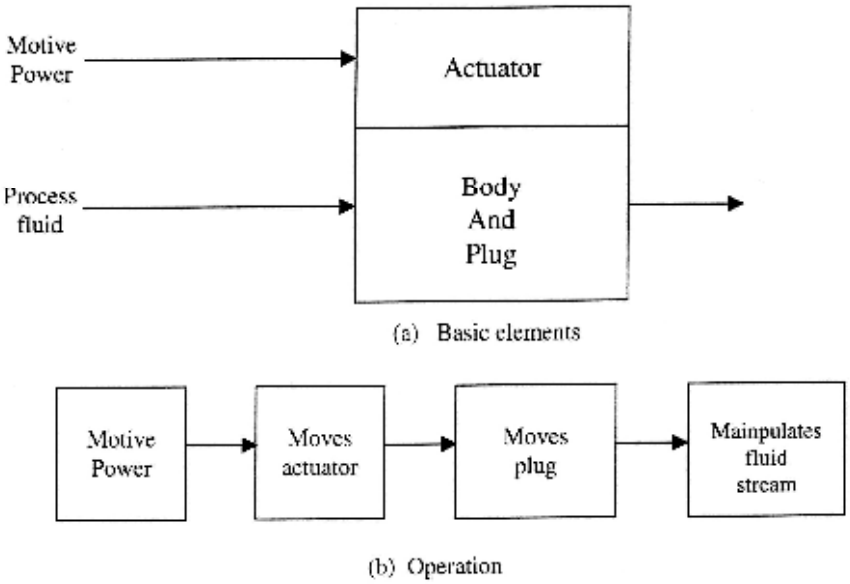


Figure 48
Control Valve Mechanism

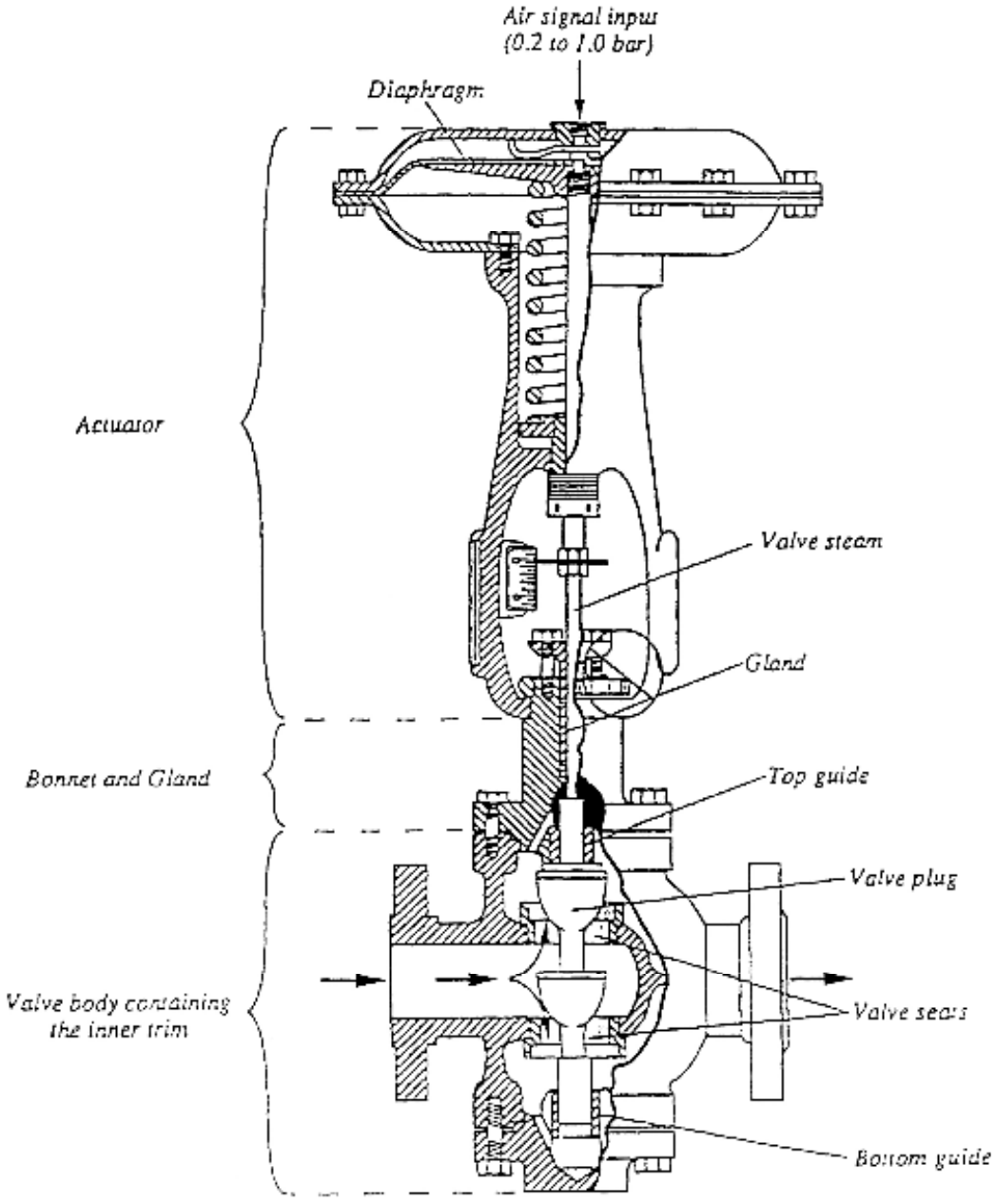


Figure 49
Components of a Control Valve

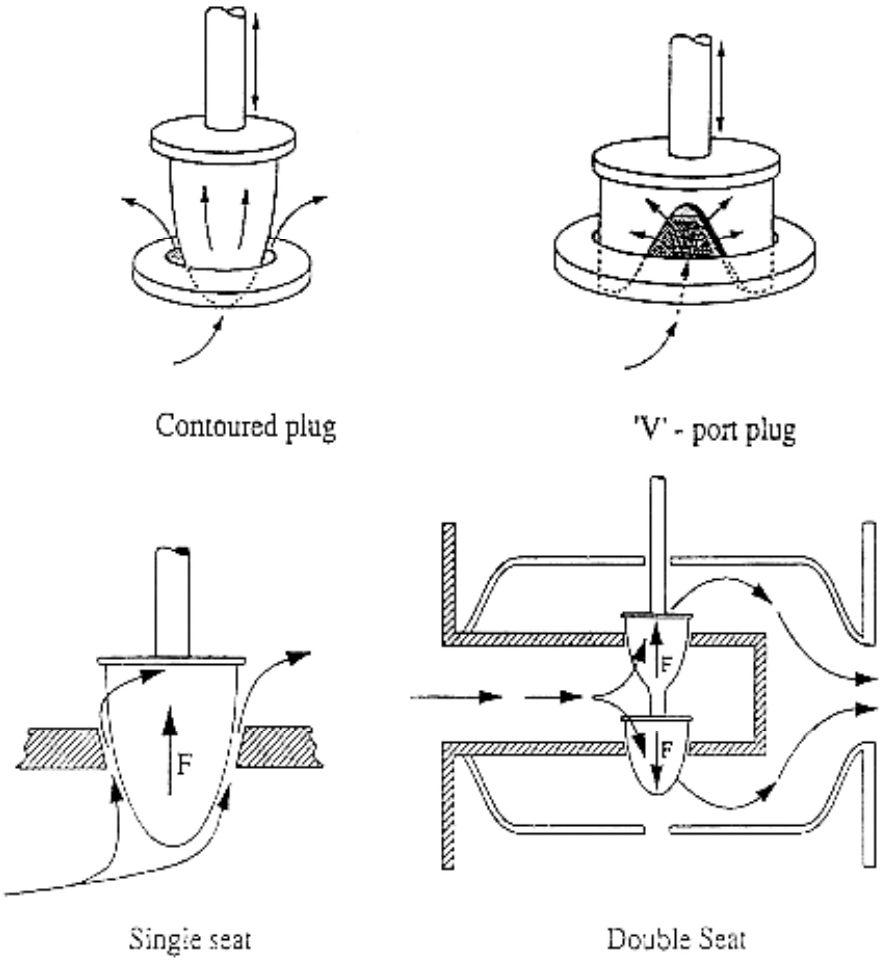


Figure 50
Types of Control Valve Trim

Control Valve Sizing

Normally a valve is designed to handle its maximum flow when it is at 75% open. Making the valve too big or too small would be detrimental to the operation of the valve and the loop. Valves should not operate below the 10% open position or above the 90% open position. When selecting a control valve, the following factors should be considered.

Suitable flow characteristics to match the process
Fail safe mode
Proper choice of valve body type and accessories.
Correct installation

Valve shutoff
There are six classes of valve leakage

Class I: no test

Class II : 0.5% of rated valve capacity

Class III : 0.1 % of rated valve capacity

Class IV: 0.0 1 % of rated valve capacity

Class V :0.005% ml/min of water per inch of port diameter per psi differential

Class VI :bubble tight
If tight shutoff is required, it is good practice to provide a tight shutoff isolation valve in series with the throttling valve. The soft seat in a throttling valve will need to be frequently replaced if it is used to carry out tight shutoff.

Valve Trim
Valves control the rate of flow by introducing a pressure drop across the valve trim. The valve trim consists of the plug and seat (these are usually sold as matched sets, which have been ground to a precise fit in the fully closed position).

All valves have a throttling action that causes a reduction in pressure. If the pressure increases again too rapidly air bubbles, entrained in the fluid implode, causing rapid erosion of the valve plug and seat surfaces. This process is known as cavitations.

Valve Characteristics

The flow characteristics of a valve defines the slow behavior of the valve as it is operated through its rated stroke. It is usually shown as a plot of valve opening or lift (% of maximum) against flow (% of maximum). The characteristics quoted by the manufacturers are theoretical or inherent flow characteristics obtained for a constant pressure drop across the valve. The actual or installed characteristics are different from the inherent characteristics, since they incorporate the effects of line losses acting in series with the pressure drop across the valve.
The three main types of characteristics are:

Quick-opening

Linear

Equal-percentage

Quick-Opening

This is the characteristic of the level seated disc or plain flat disc type of valve plug. With this type of valve, near maximum flow rate occurs at relatively small stem travel, approximately 35%, after this value there is little change in flow rate.

Linear

The linear characteristic shows that the relationship between valve opening and *flow* at constant pressure drop is a straight line, i.e. equal increments of lift should cause equal increments of flow (30% flow at 30% open, 70% *flow* at 70% open).

Equal Percentage

As the name implies, an equal percentage valve is one in which equal increments of lift cause equal percentage increments in flow. In practical terms this means that the valve will give accurate throttling control over its lower range and rapidly increasing capacity as it approaches the wide open limit. This implies that the sensitivity of the valve increases with valve opening. These valves are used when major changes in pressure occur across the valve and where there is limited data regarding flow conditions in the system.

Use of Valve Characteristics.

Normally the choice of valve characteristic required by the loop is established by carrying out a dynamic analysis of the control loop, but there are rules of thumb that can be applied to general situations.

Linear trims are used in situations, such as level control, where the pressure drop across the valve is constant.

Equal percentage trims are best used in flow situations where the pressure drop across the valve will vary as the flow goes from its minimum value to its maximum value. This is especially true on pumped systems.

Quick opening valves are useful in by-pass or re-cycle lines where a basic on/off control of flow is required.

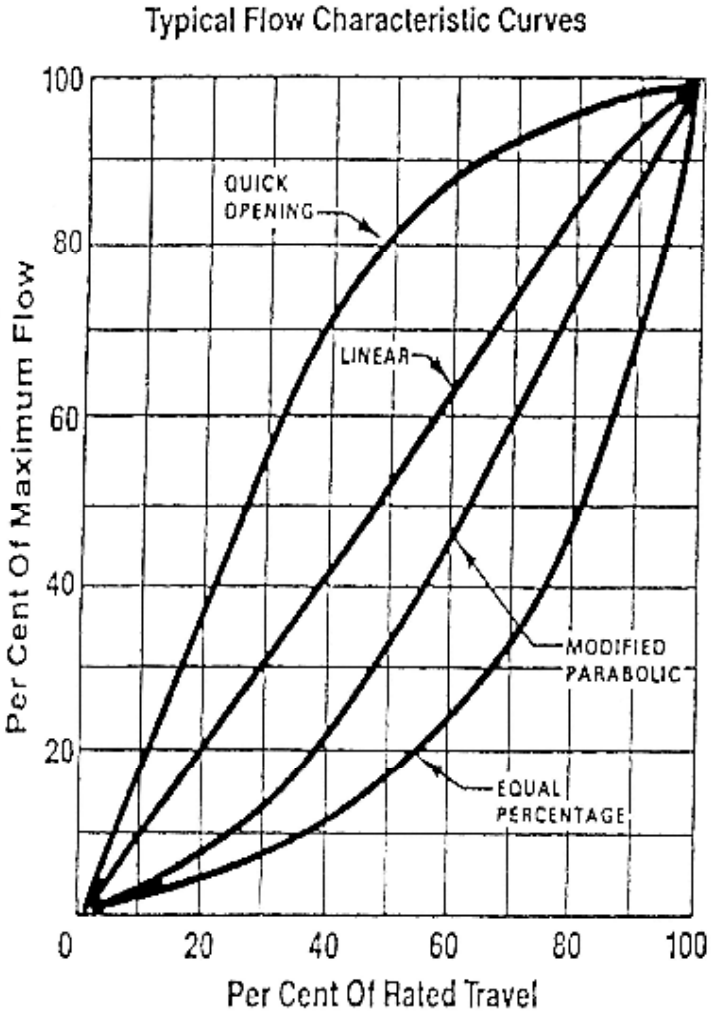


Figure 51
Typical flow characteristic curves

Valve Actuators

Actuator Types

The actuator provides the power to vary the orifice area of the valve in response to a signal received.

An actuator is classified according to its input signal and its output to the power element (valve). It maybe mechanical, pneumatic, electro-hydraulic, or electro-pneumatic.

It can receive its operating air signal from a pneumatic controller either directly or through an auxiliary device such as a valve positioner. The input signal usually varies between 0.2 and 1 Bar, acting on the effective area of the actuator. This results in a force which is used to position the control valve.

The diaphragm type of pneumatic actuator consists of a flexible diaphragm in a pressure tight housing attached to the valve stem. The diaphragm acts as a pressure seal between the upper and lower chambers. The conventional type diaphragm consists of a flat piece of flexible material of circular shape and is pre-moulded to prevent crimping or stretching. The upper and lower cases of the diaphragm housing are usually held together by bolting or clamping.

The diaphragm actuator is available with either a straight stem or a lever for connection to the final control element. These are referred to as thrust and lever type respectively. The thrust type is used where the actuator stem can be directly coupled to the valve stem such as with globe or gate valves. The lever type is used with final control elements such as butterfly valves.

Methods of Actuation

By use of the correct failure mode of the actuator, the valve can be made to **FAIL**

OPEN or **FAIL CLOSED** depending on the fail-safe requirements of the process.
The main actuator types available are:

Spring Type Diaphragm Actuator

In this type of operator, a spring opposes the force of the air pressure acting over the effective area of the diaphragm. It provides a given position of the valve stem for a specific input signal pressure received from the controller. This design is inherently fail safe, reliable and has few moving parts.

It has however limited power, limited seat shutoff capabilities and is slow to operate.

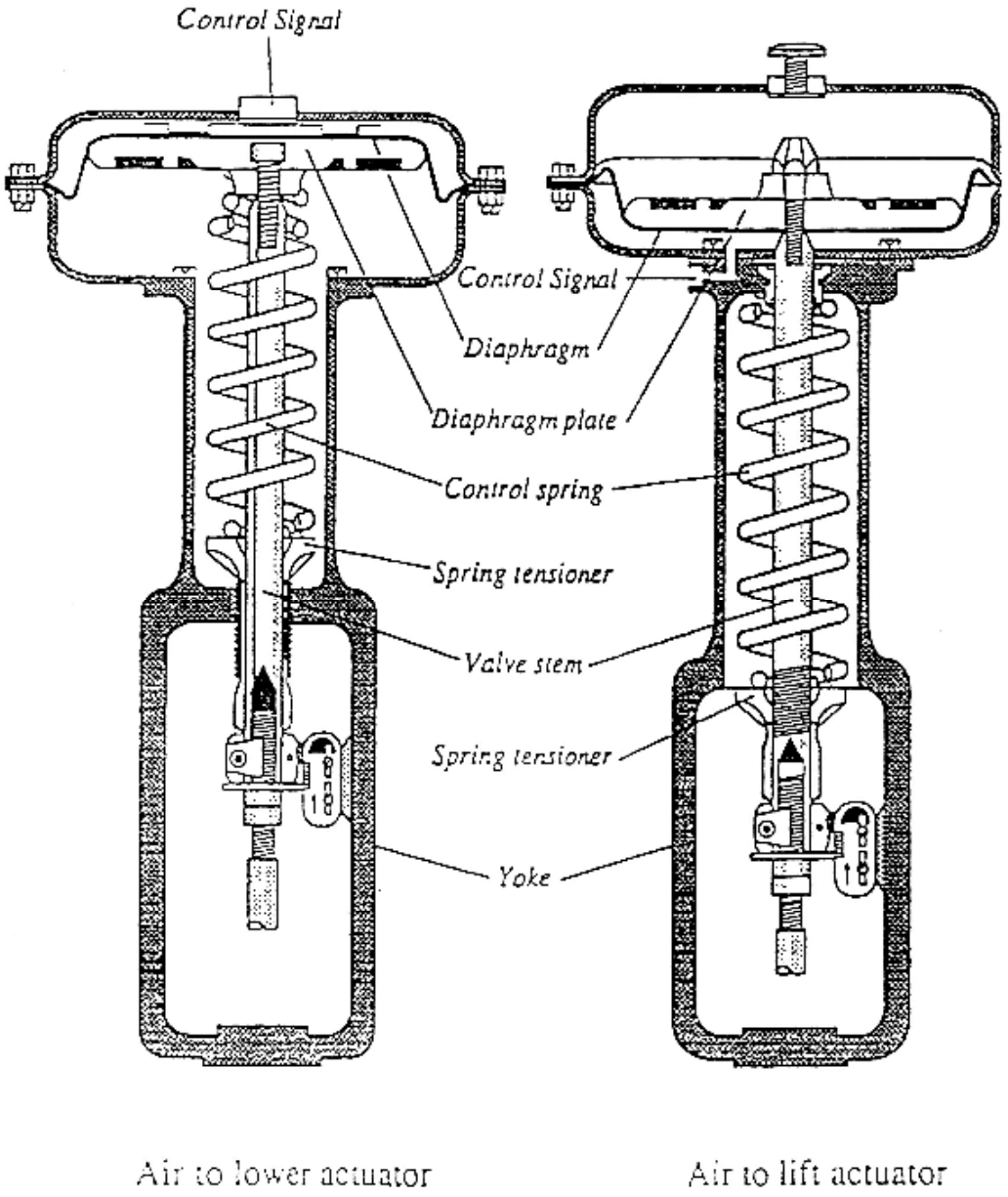


Figure 52
Diagram Actuator

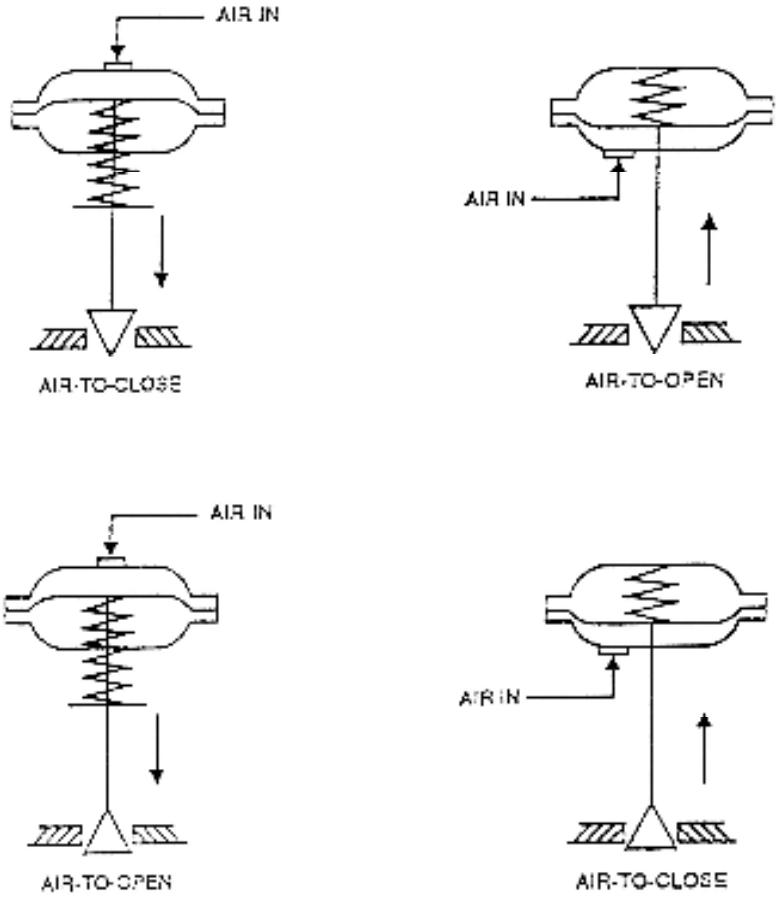


Figure 53
Valve Failure Mode

Piston Type Actuator

The air piston provides high torque or force and has a fast stroking speed. It provides a high power to weight ratio, has few moving parts and an excellent dynamic response. It can handle high differential pressures and provides high shutoff capability.

Valve Positioners

In theory, a control valve should respond quickly to small output changes from the controller. However, if the controller output pressure is very small, the actuator may not be able to develop sufficient force to position the valve correctly and/or fast enough for good control. This failure could be caused by friction between the stem and the valve packing, unbalance of the valve plug due to the hydrostatic forces of the process fluid within the valve itself.
This creates two main problems:

It takes a greater force to initially move the valve in any direction, so causing a dead spot.

Once the valve is moving, the initial force applied will cause the valve to accelerate and this in turn could cause overshoot and instability in the process

Therefore, instead of sending the signal from the controller directly to the valve diaphragm, the signal is passed to a slave controller, with its own air supply, known as a positioner.

A positioner is fitted to a valve in such a way that it can monitor the valve position and adjust its output signal until the valve is at the position required i.e. the inclusion of a positioner provides closed loop control of the valve position.
Positioners can provide the following benefits

accurate positioning of the valve stern

ability to change the valve characteristics

split the operating range of two or more valves

increase the speed of response

reverse the action of a valve

Solenoid Actuators

These are only used on small control systems where on-off control is required. Mostly they are found in the form of three way valves on the signal lines from the controller to the valve for ESD use. On removal of the power the valve will disconnect the controller from the valve and vent the air in the valve to atmosphere.

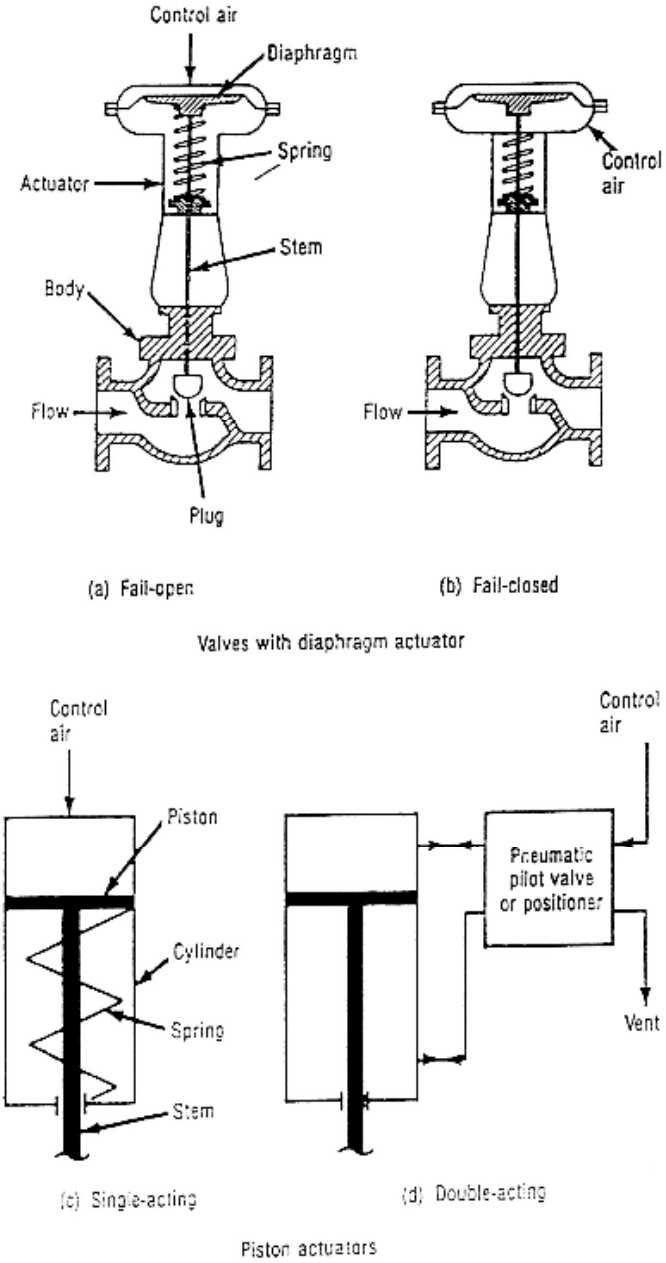


Figure 54
Diagram and Piston Actuators

Pressure Relief Valves

Relief valves operate when the pressure in a system or a vessel rises above the safe limit.

Three types of pressure-relief valves are used:

safety valves

relief valves

safety-relief valves

All three types of PZVs perform the same basic function. They relieve excess pressure in a system. Most PZVs relieve pressure to the blow-down system. Some relieve pressure to the atmosphere. This occurs when PZVs are at the top of Very tall columns, or to discharge air or steam.
In the plant, pumps, boilers, heat exchangers and vessels operate under pressure. These can operate safely a little above normal operating pressures. However, there is a limit to how much pressure a piece of equipment can withstand. Relief valves can safely vent excess pressure.

Safety Valves

High-pressure steam or gas systems use safety valves. A safety valve is a pressure relieving device. Vapors can be compressed, so safety valves must open quickly when the set pressure is exceeded. A safety valve makes a sharp popping sound when it opens.

Major Parts of a Safety Valve

Figure 55 shows the major parts of a typical safety valve. A safety valve operates automatically by static gas or vapor pressure upstream of the valve. Each valve has a hand lever so you can also operate it manually.

How Safety Valves Function

Gas pressure upstream of the valve presses on the disc. The disc connects to a spring actuator. At the set pressure, the spring starts to compress and the valve pops open.

Relief Valves

A relief valve is another kind of automatic pressure relieving device. Relief valves relieve liquid pressures. Liquids cannot be compressed. So releasing a small amount

of liquid brings the pressure down to safe levels. A typical relief valve relieves pressure at only 10% open.

Figure 56 shows a relief valve. The main parts are similar to those of a safety valve. Relief valves do not pop like safety valves. They open gradually as the upstream pressure increases.

Safety-Relief Valves

You will see safety-relief valves on all process equipment. These valves can operate either as safety valves on gas or vapor systems, or as relief valves on liquid systems. The main parts are similar to a safety valve or a relief valve. Safety-relief valves can operate as either type.

Rupture Disks

Rupture disc are installed on high pressure vessels. A rupture disc bursts in case the pressure exceeds the maximum allowable working pressure and the safety valve fails to open.

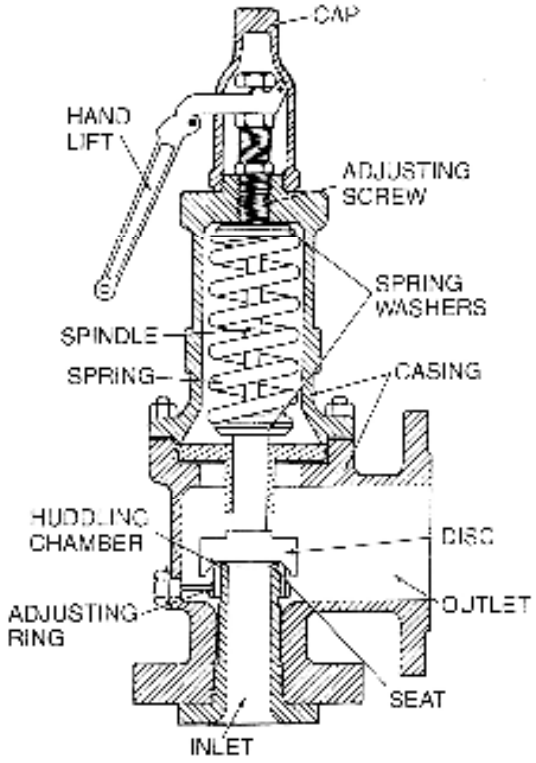


Figure 55
Spring-Loaded Safety Valve

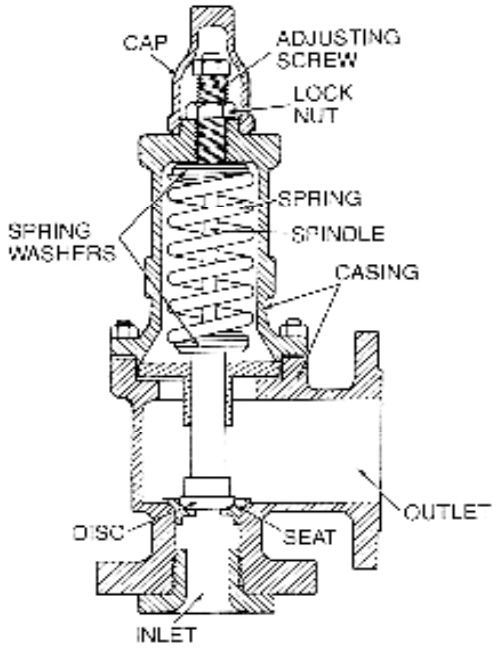


Figure 56
Spring-Loaded Relief Valve

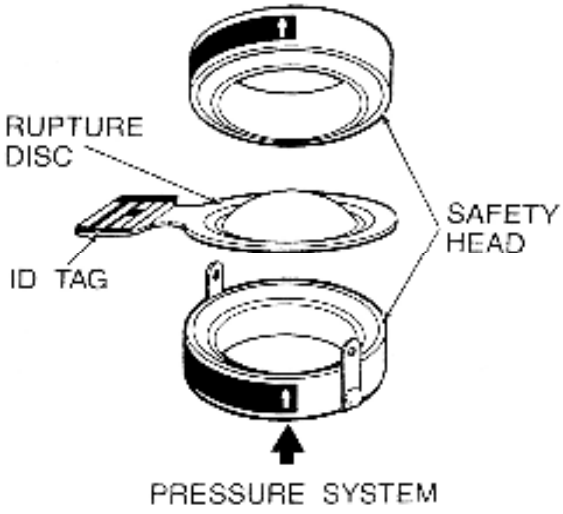


Figure 57
Tension Loaded Disc

PROCESS SAFETY SYSTEM

Safety Analysis

The API developed a document for process safety analysis (RP14 C)
In this document ten different process components have been analyzed and a Safety Analysis Table (SAT) has been developed for each component. A sample SAT for a pressure vessel is shown in Table 1.

Table 1
Safety Analysis Table (SAT)
Pressure Vessels

Undesirable Event	Cause	Detectable Condition at Component	Protection	
			Primary	Secondary
Overpressure	Inflow exceeds outflow Thermal expansion Blocked outlet	High pressure	PSH	PSV
Under pressure	Withdrawals exceed inflow Thermal contraction	Low pressure	Gas Make-up System	PSL
Overflow	Liquid inflow exceeds liquid output capacity Level control failure	High liquid level	LSH	*LSH and *PSH
Gas blow by	Level control failure	Low liquid level	LSL	*PSH and *PSV or *Vents
Leak	Deterioration Rupture Accident	Low pressure and backflow	PSL and FSV	ESS
Excess temperature (Process)	Excess heat input	High temperature (Process)	TSH	Safety Devices on Heat Source

Courtesy of API RP 14C

Emergency shut-down systems

The safety system should include features to minimize damage by shutting down appropriate equipment processes when the control system fails to release process upsets.

In addition, manual shut-down stations should be installed at strategic locations to enable personnel to shut down the production facility in case of emergency. These ESD stations should be well marked and located conveniently (50-100 ft) from protected equipment.

Annunciation systems

These systems give early warning of impending trouble to allow personnel to take corrective action prior to a shut in, and provide information about the initial cause of a shut-in. They are a vital part of any large shut-down system design. On smaller systems, process alarms may be minimal as there may not be sufficient time for personnel to react to the alarm before an automatic shut-down is initiated.

Annunciator panels should be at a central location with alarm annunciators and shut-down annunciators grouped separately. The first alarm and the first shut-down normally sound a horn and are annunciated. This is called "first out indication." Subsequent shut-down or alarm signals received by the panel are either not annunciated or are annunciated in a different manner so that the operator can determine the initiating cause of the process upset.

Alarm signals may come from the output signal used to control an operational valve. Shut-down signals should come from a completely separate instrument not dependent upon a normally used output signal for operation.

Function Charts

Figure 58 is a simple process flow diagram. The function chart for this simple process flow diagram is shown in figure 59.
In a function chart each sensing device is listed on the left hand side and a path is then drawn showing the route of the signal from the sensing device to the device that performs the shut down or alarm function.

Safety Device symbols

Table 2 shows symbols used in RP 14 C to represent the various sensors and shut-down devices.

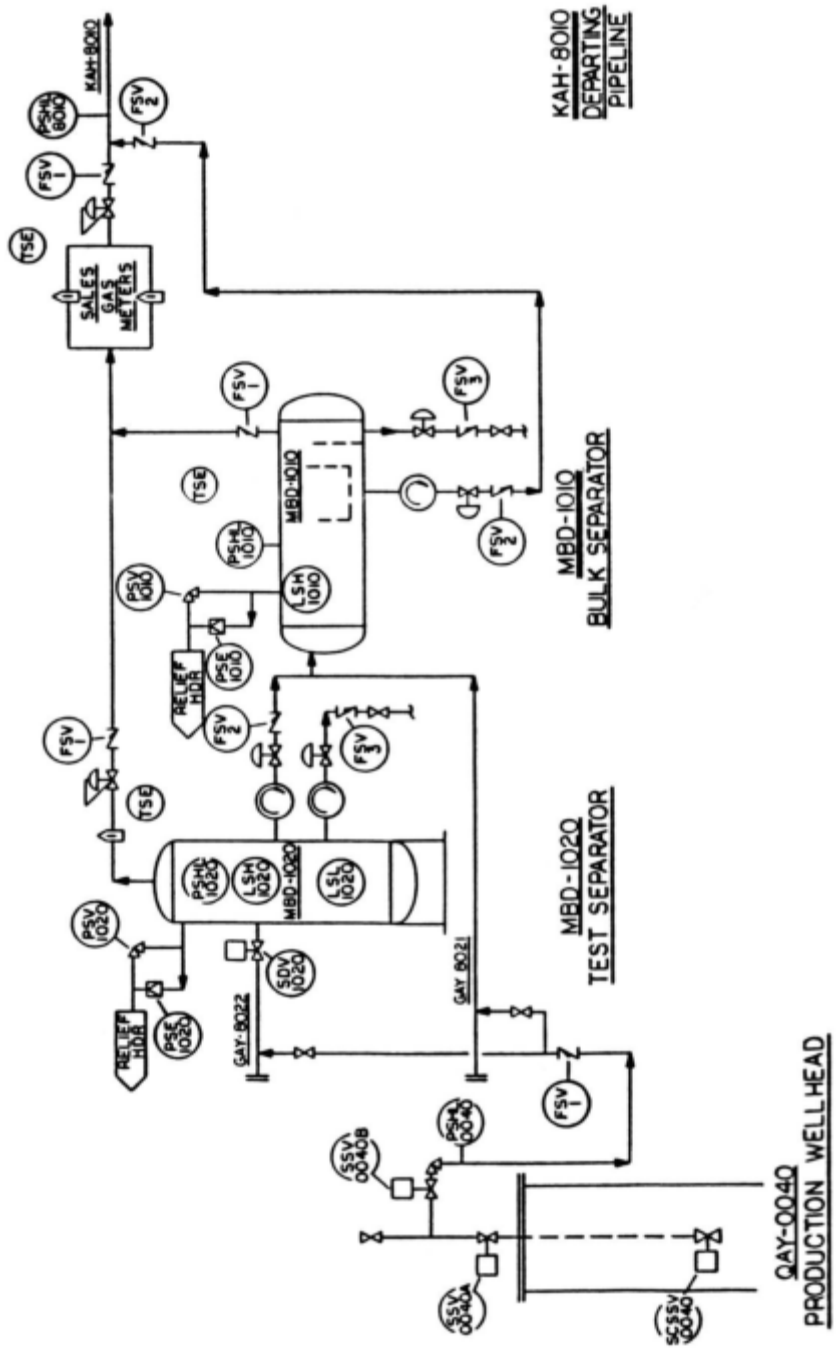


Figure 58
Simple Process Flow Diagram.

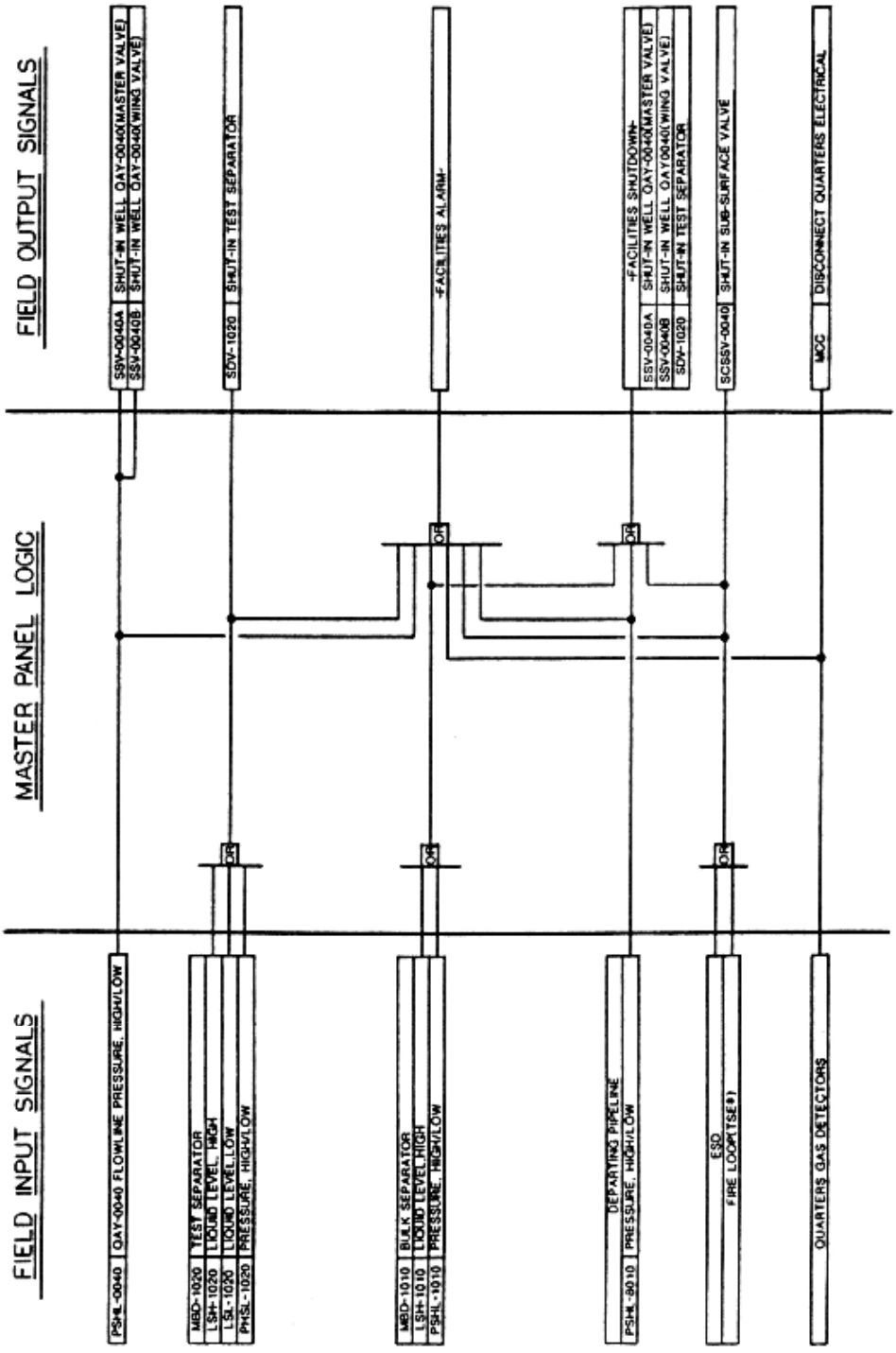
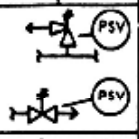
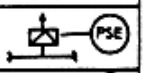
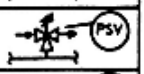
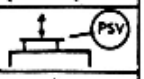
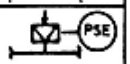
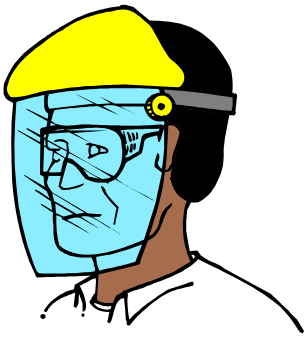


Figure 59
Function Chart

Table 2
Safety Device Symbols

SENSING AND SELF-ACTING DEVICES				
VARIABLE	SAFETY DEVICE DESIGNATION		SYMBOL	
	COMMON	INSTRUMENT SOCIETY OF AMERICA (U.S.A.)	SINGLE DEVICE	COMBINATION DEVICE
BACKFLOW	CHECK VALVE	FLOW SAFETY VALVE		
BURNER FLAME	BURNER FLAME DETECTOR	BURNER SAFETY LOW	BSL	
COMBUSTIBLE GAS CONCENTRATION	COMBUSTIBLE GAS DETECTOR	ANALYZER SAFETY HIGH	ASH	
FLOW	HIGH FLOW SENSOR	FLOW SAFETY HIGH	FSH	FSHL
	LOW FLOW SENSOR	FLOW SAFETY LOW	FSL	
LEVEL	HIGH LEVEL SENSOR	LEVEL SAFETY HIGH	LSH	LSHL
	LOW LEVEL SENSOR	LEVEL SAFETY LOW	LSL	
PRESSURE	HIGH PRESSURE SENSOR	PRESSURE SAFETY HIGH	PSH	PSHL
	LOW PRESSURE SENSOR	PRESSURE SAFETY LOW	PSL	
	PRESSURE RELIEF OR SAFETY VALVE	PRESSURE SAFETY VALVE		
	RUPTURE DISC OR SAFETY HEAD	PRESSURE SAFETY ELEMENT		
PRESSURE OR VACUUM	PRESSURE - VACUUM RELIEF VALVE	PRESSURE SAFETY VALVE		
	PRESSURE - VACUUM RELIEF MANHOLE COVER	PRESSURE SAFETY VALVE		
	VENT	NONE		
VACUUM	VACUUM RELIEF VALVE	PRESSURE SAFETY VALVE		
	RUPTURE DISC OR SAFETY HEAD	PRESSURE SAFETY ELEMENT		
TEMPERATURE	FUSIBLE MATERIAL	TEMPERATURE SAFETY ELEMENT	TSE	
	HIGH TEMPERATURE SENSOR	TEMPERATURE SAFETY HIGH	TSH	TSHL
	LOW TEMPERATURE SENSOR	TEMPERATURE SAFETY LOW	TSL	
FLAME	FLAME OR STACK ARRESTOR	NONE		

HSE



INTRODUCTION

- Because safety is so important in Petroleum Companies, it will be the first subject in this orientation.
- Remember there are safe jobs that are made dangerous by being careless. There are dangerous jobs that are done safely by using the correct procedures and the right equipment.
- You will learn about personal safety clothing and equipment. Also you will learn about the safety equipment that is provided by Petroleum Companies for your safety. However you will also learn about some of the hazards that may be found in any plant at Petroleum Companies.
- Every industrial plant has built-in potential safety hazards; Notice the use of the word potential. This means something that is possible, but hasn't happened yet. There is no need for potential safety hazards to become an accident if you know what the hazards are and how to guard against them, you can avoid accidents.

PERSONAL SAFETY CLOTHING AND EQUIPMENT

- Safety on the job is very important. It is important to you and to Petroleum Companies. The Company does everything it can to make its plants safe places to work.
- Petroleum Companies has safety rules that must be followed. Remember that no one can make you a safe worker except yourself. You must practice safety at all times.
- You have been issued with personal safety clothing and equipment. It is yours and you will use it wherever you go in SGC. Each item is designed to protect you against certain hazards you will be exposed to in your work. Each item protects a vital part of your body.

IMPORTANT

- Wear the correct safety clothing in training Center and on the job. Your life May depend on it!
- Let's look at each article of clothing and equipment and see:
  - What it is made of.
  - What it protects you against.
  - How to use it correctly.

SAFETY SHOES

- Safety shoes are made of good quality leather. The soles are made of rubber. Check the soles of your safety shoes to make sure they have a good tread. Rubber soles reduce your chances of slipping. They are made to withstand damage from oils and greases.
- Inside the shoe there is a steel toe-cap. If a heavy or pointed object falls on your foot, the steel toe-cap will protect your toes from being crushed.
- Look after your safety shoes. Do not damage the backs. Always tie the laces correctly. You can trip over loose laces.

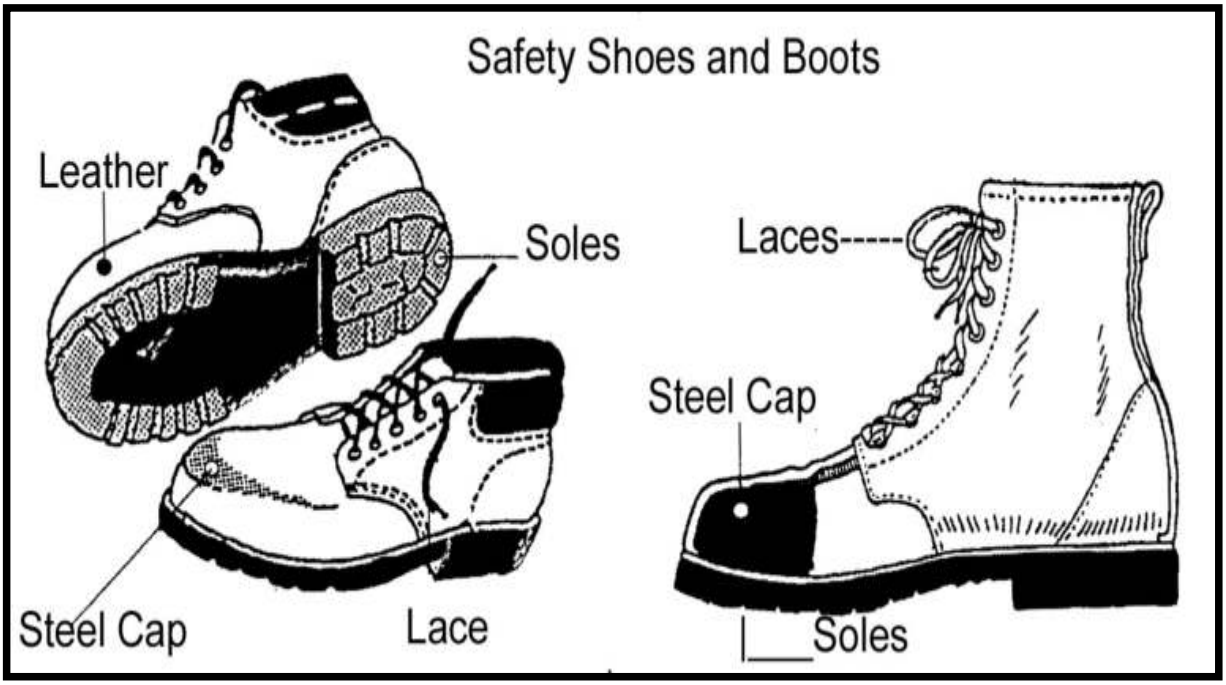


Figure 1: Safety Shoes

Important

Safety shoes must be worn at all times in any industrial area in SGC.

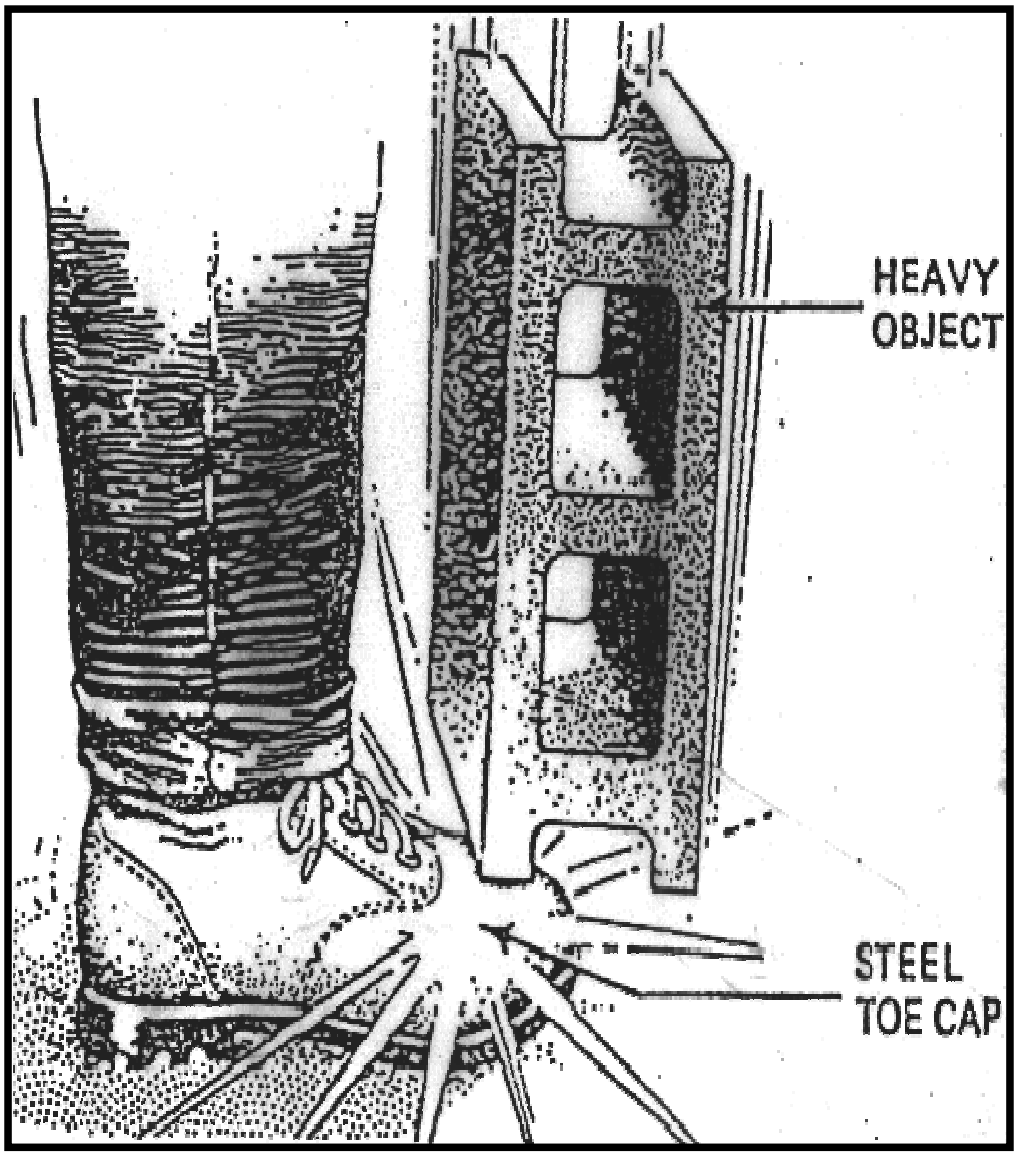


Figure 2: Toe Protection

SAFETY GLASSES

- Safety glasses are made with shatterproof lenses and have plastic shields at the sides. Do not use safety glasses without side shields. Side shields are to protect your eyes from objects coming from outside your line of vision.

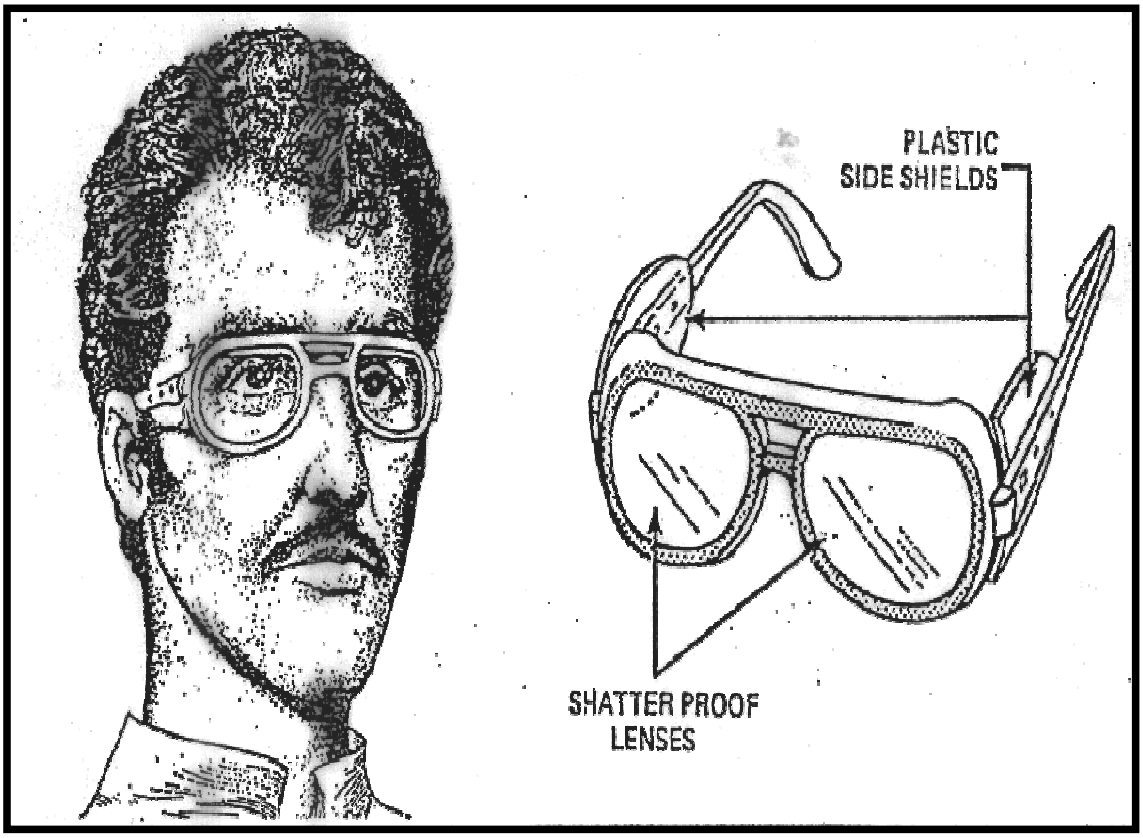


Figure 3: Safety Glasses

- SGC will issue prescription safety glasses.
- The plastic arms of the glasses can be made to fit the wearer by :
  - Placing the arm in hot water to soften the plastic.
  - Bending the arm to the shape of your head.
- Be sure your safety glasses fit tightly so that they won't fall of when you bend over.

Important

In SGC, you must wear safety glasses in workshops and in all outside plant areas.

HARD HAT

- Hard hats (safety helmets, safety hats) are made of rigid plastic. Do not paint them because it makes them brittle.

Figure 4: SAFETY HAT or HELMET

- They are designed to protect your head from injury. To do the job properly, the hard hat must be worn as shown in the pictures below:

Figure 5: Correct wearing of a Safety Helmet.

Safety Helmets are to protect you from serious accidents. Always wear the helmet correctly. You may have to wear the helmet for a long time. Do not put Baseball Caps underneath them. Make sure the helmet is fitted well and not too tightly. Adjust the size of the helmet if necessary. Make sure the helmet is protecting the sides and front of your head too. Do not wear the helmet at an angle too far back on your head.

- Hard hats must always be worn in an industrial site or when there is danger from overhead.
- In your work as a plant operator, you will frequently walk under low pipes and platforms.

Important

You must wear your hard hat in all outside plant areas.

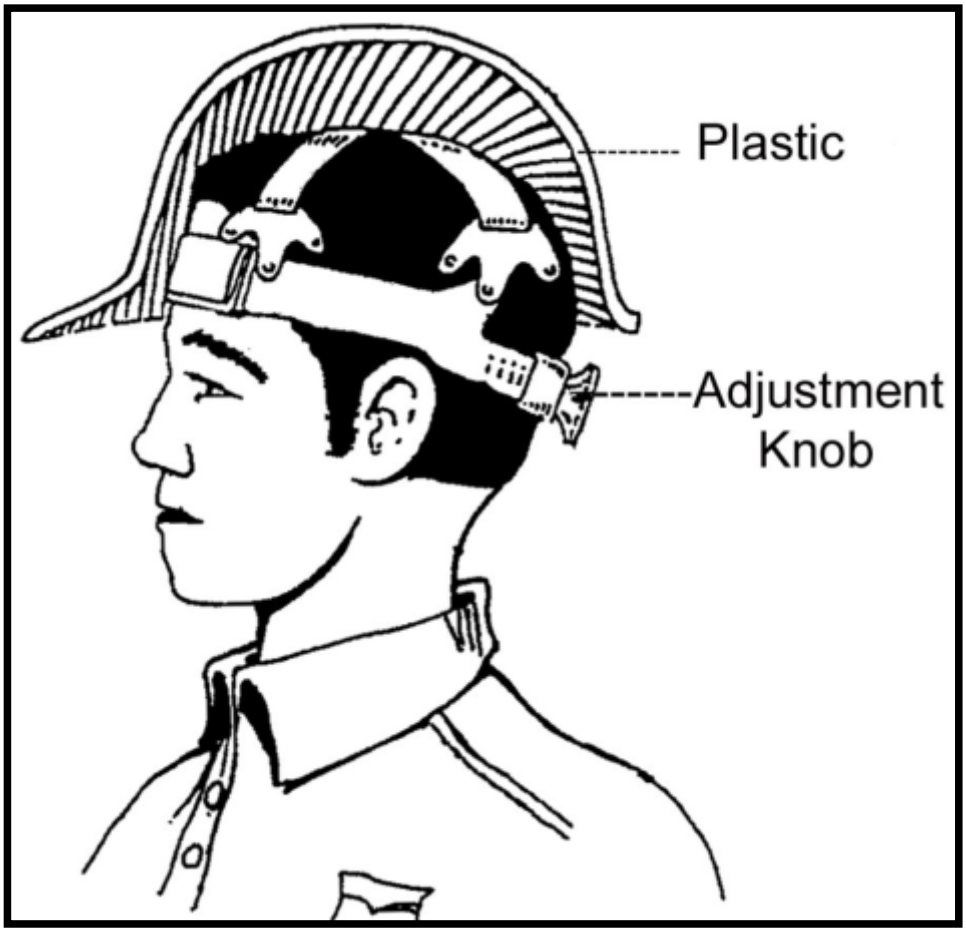


Figure 6: Hard Hat Adjustment

Important

Your hard hat can save you from a serious head injury.

EAR MUFFS/EAR PLUGS

- Continuous, loud noise can cause a loss of hearing. Some parts of SGC plants are very noisy. The noise is loud enough to damage your hearing. If you must work in noisy areas for long periods of time, you should wear ear muffs.
- Ear muffs attach to your safety helmet. They can be turned back off your ears when you do not need them.
- Ear plugs, worn under ear muffs, provide even more ear protection.
- Ear plugs, are available for use in Petroleum Companies plants.

Important

Some work areas on the job may be subject to loud noises, for example, when new construction work is performed. Wear earmuffs and earplugs to protect your hearing in these areas.

Figure 7: Ear Plugs

Figure 8: There are many different types of Ear Muffs and Plugs

GLOVES

- In hot places, metal objects that are out in the sun get hot. Very hot! A painful burn can result when you torch sun-heated metal.
- There are objects with rough surface. Some may have sharp edges.
- To protect hands against these hazards, SGC has issued you gloves.
- Safety gloves have leather palms. The cuffs and backs are made of heavy cotton.
- Occasionally, you will have to do work that requires a sense of touch. You will need to remove your gloves. They are not made for delicate work.

Important

Wear Safety gloves when operating equipment located outside the plant.

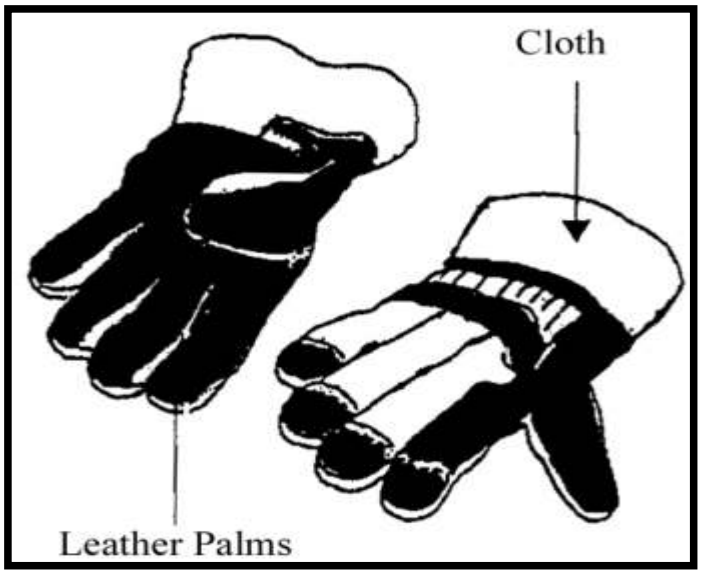


Figure 9: Safety Gloves

SAFETY SUMMARY

- There are no spare parts for you eyes. Wear your safety glasses.
- Your hands are your most valuable tools. Protect them. Wear your gloves.
- Assume that every heavy object that falls will land on your toes. Wear your safety shoes.
- If there is any chance of something hitting you on your head, wear your hard hat.
- Hearing loss from continuous loud noise is permanent. Wear earmuffs and earplugs.

PLANT/ TRAINING FACILITY and WORKSHOP/ CLASSROOM SAFETY EQUIPMENT

- Each Plant/workshop/classroom has safety equipment for your protection. You need to know where these items of equipment are located.

Fire Extinguishers

- Every Plant/workshop will have one or more fire extinguishers, which are used for fighting different types of fire, for farther details see pages 67, 68 & 69.

Figure 10: Fire Extinguisher

- Petroleum Companies personnel have been trained to fight fires. They know how to operate fire-fighting equipment.
- If you see a fire, you should certainly give the alarm.
- **Your responsibility is to evacuate the plant/building in a prompt and orderly way.**

Caution

- Do not handle or operate fire equipment when an alarm sounds. Evacuate the plant/building and take the safe action according to your role at HSE Policy.

FIRE BLANKET

- A fire blanket is made of heavy non-flammable material. The purpose of a fire blanket is to wrap around a person if his clothing is on fire.
- The fire blanket itself will not burn. The blanket will smother the flames by cutting off air that the fire needs in order to burn.

Important

- **The fire blanket is made of special material that will not burn.**
- **Use the blanket to wrap around a person whose clothes are on fire.**

Figure 11: Fire Blanket

EYE BATH

- The eye bath supplies potable water from two spray nozzles. The nozzles are aimed so that the water will spray into your eyes when you place your face close to the bath.
- The eye bath is intended mostly for washing chemicals and acids that may be splashed into your eyes.
- The eye bath can be used to wash out particles of dirt that may get into your eyes.
- **The water spray pressure is not strong enough to damage your eyes.**

Important

- **You should use an eye bath to wash your eyes if acid or chemicals ever splash in your face. Find out where the eye baths are located in your work site.**

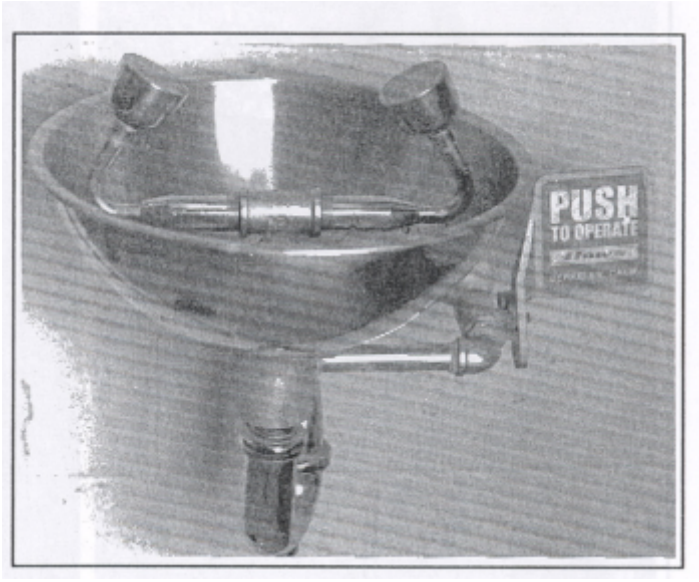


Figure 12: Eye Bath

- Wherever you go in Petroleum Companies, you will hear about safety. The plant to which you are assigned will have its own special safety rules and procedures. Even though you may work safety, others around you may not. Be alert to what is happening around you.

- It is not possible to identify every safety hazard or every unsafe act. This module will show you some of the more common hazards, precautions, and unsafe acts.

- This will develop your safety awareness, to get you to think safety in your work.

Figure 13: A Safe Plant is My Responsibility

IDENTIFY POTENTIAL SAFETY HAZARDS

- Built-in plant safety hazards are a basic feature of any industrial plant. There are things you can trip over. There are places you can fall from. There is rotating machinery. There are hot things.
- Several potential safety hazards are present. You have to live and work among them. The only thing you can do about them is to follow established procedures to ensure safe operations.
- Unsafe acts that are created through carelessness or lack of knowledge are different. You **can** do something about them.
- In this part of the module, we will look at some typical plant scene that will identify the following potential safety hazards:
  - Leaks
  - Falls
  - Tripping
  - Hot / Cold pipelines
  - Rotating machinery
  - Oil spills
  - Heat from the sun
  - Construction and maintenance work
  - Radiation
  - Plant / Vehicles
  - Overhead work
  - ARC welding
  - Lifting
- If you observe any unsafe act or unsafe working condition in the plant, stay calm and report to your supervisor or control room with relevant information. Make yourself safe before taking any action to control the situation.
- If you see an accident in the plant be calm and first report to your supervisor or control room with proper details before approaching to help the victim

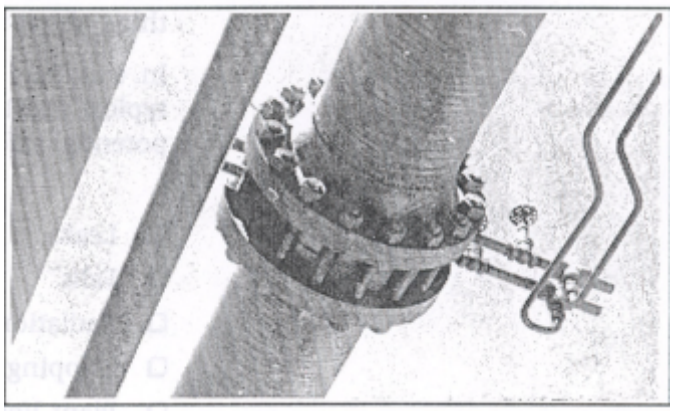
Important:

Emergency telephone No.

LEAKS

- Any place where two pipes are joined and where pipes are joined to vessels and tanks is a source of potential leaks. It isn't so much the leak that is dangerous, as what is leaking out.
- The leak could be water, or it could be a colourless liquid that looks like water but is highly corrosive, for example, acids of caustics. It could be superheated steam.
- It could be sweet crude oil, or it could be sour crude oil containing poisonous and explosive hydrogen sulphide gas. Hydrogen sulphide (H_2S) is known as a killer gas.

Figure 14: Pipe Flange Joint



- Pressure and heat inside a pipeline can eventually create a leak in a pipe joint. Vibration, caused by a large machine, can loosen pipe joint bolts. Corrosive liquids and gases can eventually eat away the gasket in a pipeline joint, and create a leak.

When you discover a leak in your plant, there are two things you must do:

- **Report the leak immediately.**
  - Report the leak and its location to your supervisor immediately. He will take the necessary correct action to handle the situation. An important reason for reporting the leak is so that other plant personal can be told about it.
  - Another important reason is that very small leaks in hydrocarbon plant pipelines can quickly become very big leaks.

- **Avoid contact with the leak substance**
  - Do not put your bare hands or fingers in the substance that is leaking.
  - Never attempt to locate any leak in a pipeline by trying feeling the leak with your hand.

H_2S GAS OR KILLER GAS

H_2S is a toxic gas and consider the most dangerous gas that causes a lot of fatality to people working in process plants having H_2S gas.

The characteristics of H_2S are:

Colour	**:**	Colourless
Smell	**:**	Rotten egg smell at low concentration of less than one PPM.
Heaviness	**:**	Heavier than ear and can accumulate in sumps, halls and excavations.
Solubility	**:**	Soluble in waters forming corrosive acid.
TLV	**:**	Threshold limit value. Below 10 PPM, at this level a Person can work for 8 hours without problem.
Flammability	**:**	Flammable in the range 4.5 to 45.5% volume in Air and burns to sulphur dioxide (SO_2), a particularly irritant gas. This is also toxic.

Table of H_2S Effects:

Concentration	Effect
About 30 ppm	Eye and respiratory tract irritation.
About 100 ppm	Loss of sense of smell.
About 500 ppm	Dizziness and headache.
1000 ppm and above	Rapid unconsciousness followed by death within minutes.

Normal Petroleum companies practice is to paint pipework and vessels continuing hazardous concentrations of H_2S Yellow or with Yellow bands and in some cases with local warning signs.

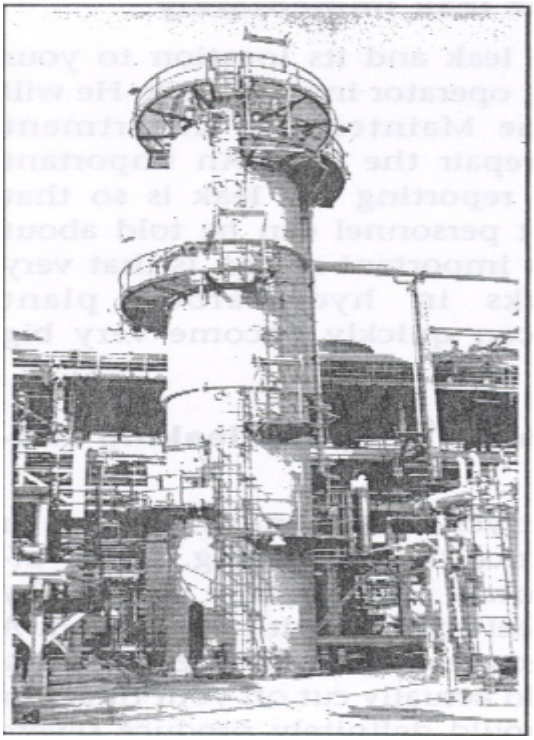
Important:

Use H_2S personnel portable monitors that provide local detection And warning of the presence of H_2S if concentration above 10ppm. Immediately stop your work and without panicking, go outside to safe areas (nearest assembly area)

FALLS

- There are many elevated platforms in Petroleum Companies plants. They are strong and they are safe. Every platform that is elevated above ground level should have a protective railing around it to prevent you from falling.
- **Elevated platforms are used because valves and instruments cannot always be located at ground level. The railing around the platform is there to keep you from falling of the platform.**
- **If you must go out side the railing use a safety belt.**

Figure 15: Elevated Platforms



Vertical Ladders

- The cage around a long vertical ladder is a safety feature that must not be by-passed. The cage will help you if you miss your grip on the next rung of the ladder and start to swing or fall backwards off the ladder. Never climb up on the outside of the safety cage.

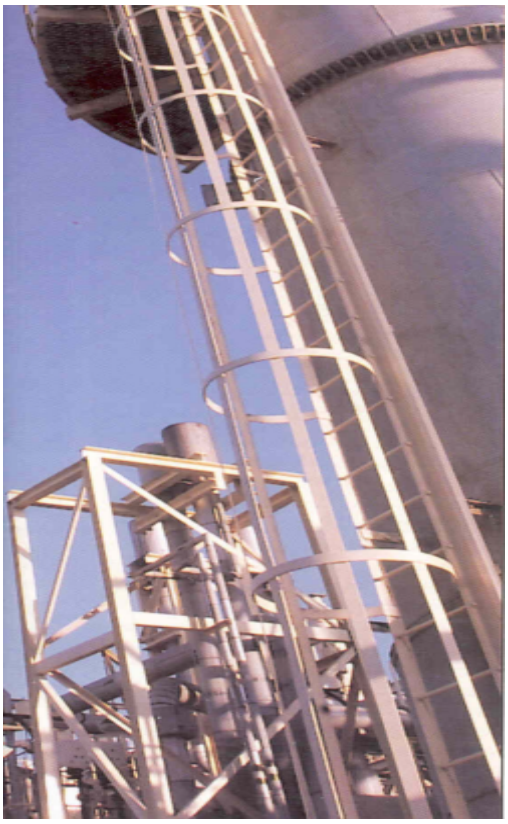


Figure 16: Ladder with Cage

Climbing Safety

- All ladders in Petroleum Companies plants are made of strong steel, but it is possible for the rungs on a ladder to break. If you find a broken rung on a ladder, report it to your supervisor. He will make sure no one uses the ladder until it has been repaired by the maintenance department.
- Use your gloves when you climb a ladder. The metal rungs and rails can be rough, or hot. On a long climb, you could injure your hands if you climb bare handed.
- Be sure that the soles of your safety shoes are free from oil or grease before you climb a ladder. Place your feet firmly on every rung as you climb.
- Before you climb up or down a ladder make sure there is no one else already on the ladder. **Make it a rule that at any one time there is only one person on a ladder. One person: One Ladder.**

Portable Ladders

- A ladder is easy to use, but accidents involving ladders can occur. Figure 17 illustrates the correct, safe way to use a ladder.

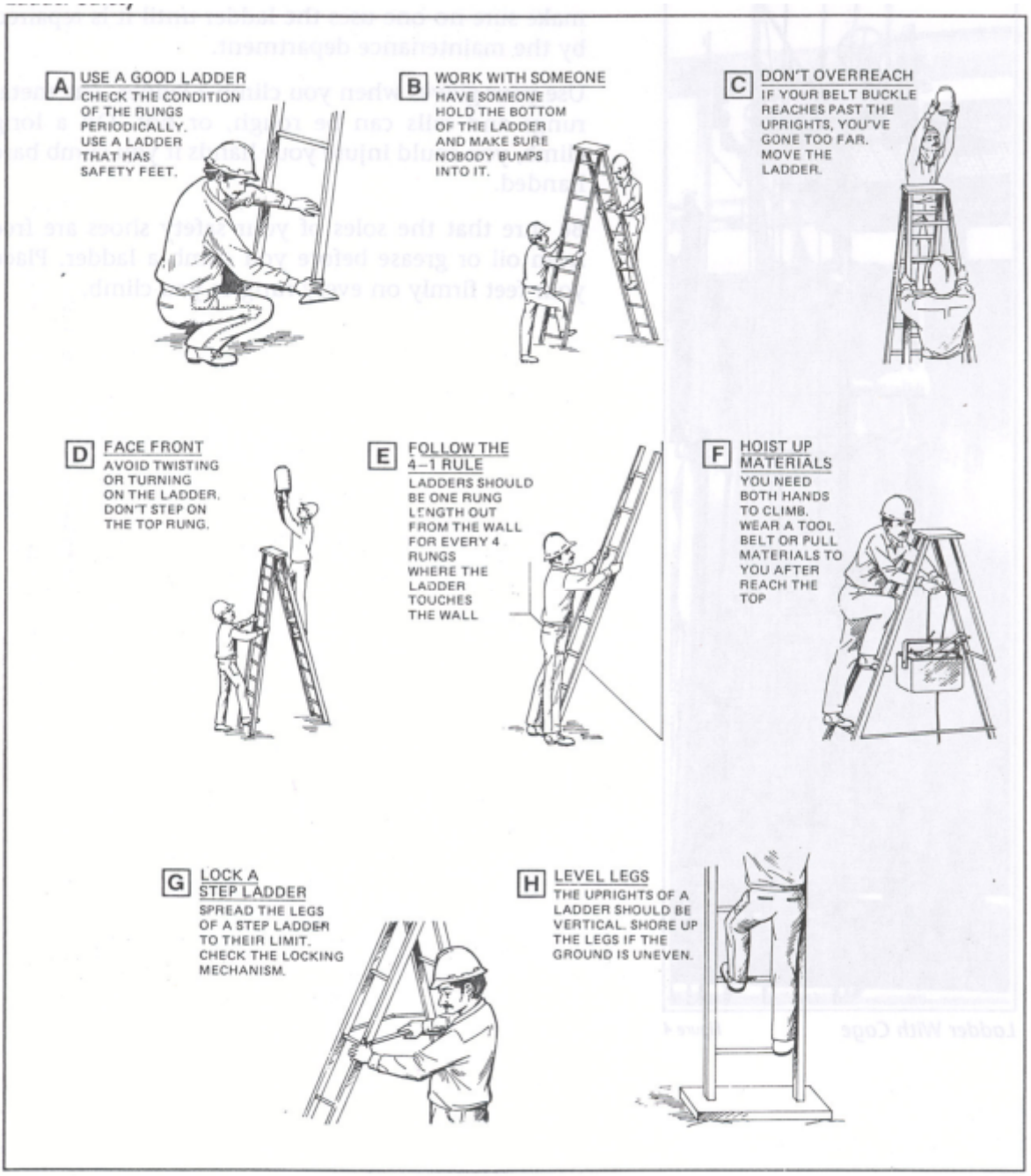
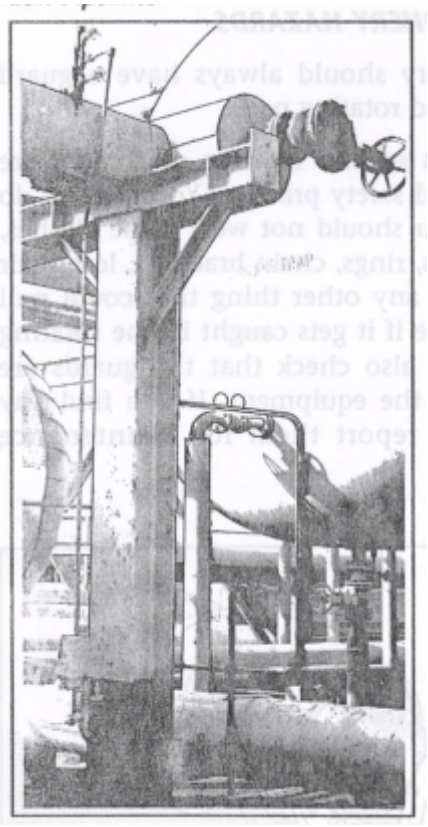

A USE A GOOD LADDER
CHECK THE CONDITION OF THE RUNGS PERIODICALLY. USE A LADDER THAT HAS SAFETY FEET.
B WORK WITH SOMEONE
HAVE SOMEONE HOLD THE BOTTOM OF THE LADDER AND MAKE SURE NOBODY BUMPS INTO IT.
C DON'T OVERREACH
IF YOUR BELT BUCKLE REACHES PAST THE UPRIGHTS, YOU'VE GONE TOO FAR. MOVE THE LADDER.
D FACE FRONT
AVOID TWISTING OR TURNING ON THE LADDER. DON'T STEP ON THE TOP RUNG.
E FOLLOW THE 4–1 RULE
LADDERS SHOULD BE ONE RUNG LENGTH OUT FROM THE WALL FOR EVERY 4 RUNGS WHERE THE LADDER TOUCHES THE WALL
F HOIST UP MATERIALS
YOU NEED BOTH HANDS TO CLIMB. WEAR A TOOL BELT OR PULL MATERIALS TO YOU AFTER REACH THE TOP
G LOCK A STEP LADDER
SPREAD THE LEGS OF A STEP LADDER TO THEIR LIMIT. CHECK THE LOCKING MECHANISM.
H LEVEL LEGS
THE UPRIGHTS OF A LADDER SHOULD BE VERTICAL. SHORE UP THE LEGS IF THE GROUND IS UNEVEN.

Figure 17: Ladder Safety

TRIPPING

- There are many low pipelines in Petroleum Companies process plants.
- Some pipelines at ground level are so big a walkway is needed to get over them. Others are small and you may be able to step over several in one step. They are the kind you trip over if you are not careful. Always watch where you are going when you move around in the plant.

Figure 18: Low Pipelines



HOT/ COLD PIPELINE HAZARDS

- Insulation on pipes helps to keep liquids flowing through them at the required temperature. Insulation protects you from very hot or very cold pipes.
- Thick insulating materials usually keep you from touching hazardous pipes, but you must be careful around the pipe flanges and valves. These may not be insulated. Also, the insulation on a pipeline may be damaged.
- Remember that you may not intend to touch a pipe, but there is always a chance you could do so accidentally. The safe thing is to always wear your personal protective clothing, such as gloves, if you work close to pipelines.
- If you find any damage to insulation in your plant, report it at once to your supervisor.

ROTATING MACHINERY HAZARDS

- Rotating machinery should always have a guard around any exposed rotating parts.
- Covers and guards around rotating machinery are only a part of good safety practice. You have to do your part also. You should not wear loose clothes, unbuttoned sleeves, rings, chain bracelets, long hair or long beards, or any other thing that could pull you into a machine if it gets caught in the rotating parts.
- You should also check that the guards are correctly fixed on the equipment. If you find any damaged guards, report them for maintenance action.

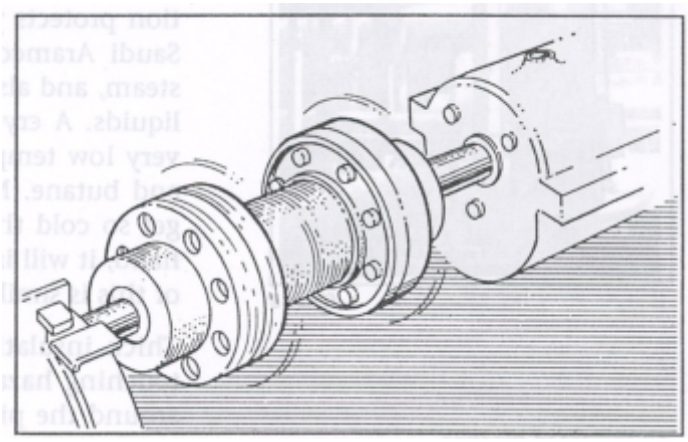


Figure 20: A Rotating Hazard

OIL SPILL HAZARDS

- Oil spills are not common in Petroleum Companies plants but they do happen. Clean up any oil spills in your area immediately. If this is not possible tape off the area to prevent other people from walking into the spill.
- Remember these things about oil spills.
  - Oil is flammable. Oil is a fuel
    Fuel + heat + oxygen = Fire
  - Watch your step. Oil is naturally slippery. If you must walk in an oil spill, go slowly and be careful.
  - Report the oil spill to your Area supervisor.

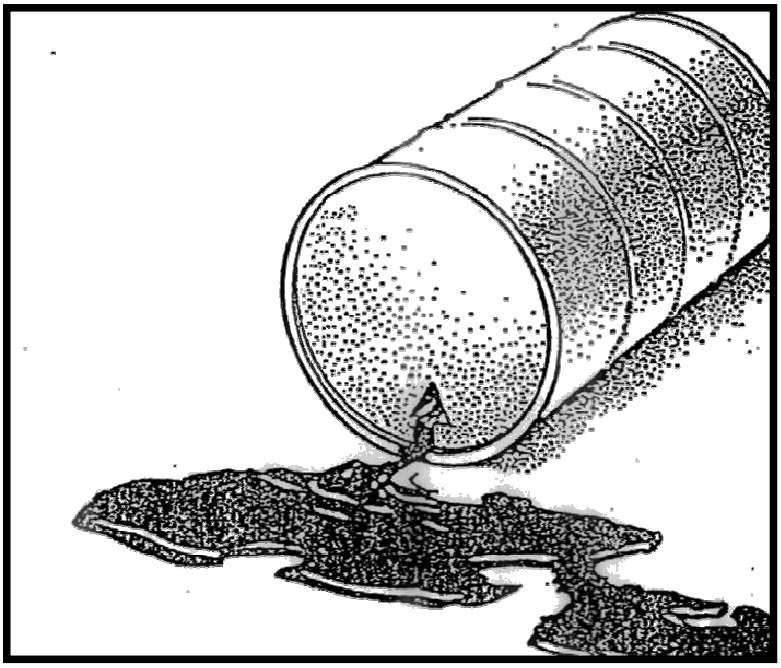


Figure 21: An Oil Spill.

HEAT HAZARDS FROM THE SUN

The sun can cause two hazards in a plant:

- Hot Objects
- Heat Exhaustion

Hot Objects

- Objects can become so hot from the sun's radiation that they can burn any part of your body that touches them. The sun can heat objects to temperatures of 150°F and higher. This is enough to burn exposed skin if the objects are touched.
- During the hot summer months, always wear your gloves when you work in the plant.

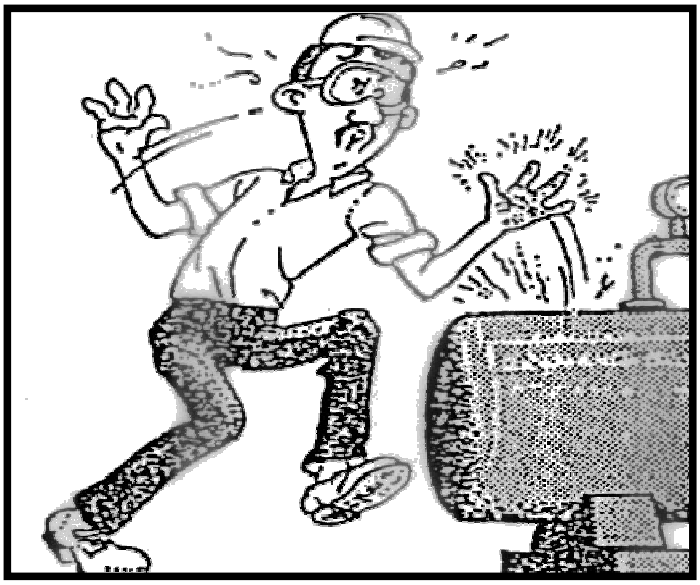


Figure 22: Too Hot to Handle

Heat Exhaustion

- Heat exhaustion is a medical condition, like an illness. It is caused by doing too much physical work in a very hot place. You become weak, and feel sick and dizzy. You usually sweat a lot.
- An extreme form of heat exhaustion is heat stroke, when sweating stops and body temperature becomes very high, leading to collapse. Heat exhaustion and heat stroke are especially dangerous when you are working above the ground or near rotating machinery, because you could fall, or get caught in machinery.
- Heat exhaustion and heat stroke can be avoided. Drink enough liquids, and try not to spend too long in direct sunlight. Your duties in a plant will normally allow this.

Figure 23: Too Long in the Sun

CONSTRUCTION AND MAINTENANCE WORK HAZARDS

- Construction and maintenance work is quite common in SGC plants. New equipment is installed. Old equipment is repaired. Underground pipes and cables are dug up for replacement and repair. The work crews that do the work should put up barriers and warning signs around their work site.

Figure 24: The Plant is at its most dangerous during Construction and Maintenance

- Unless your duties require you to enter an area where construction work is being performed, stay out of the area. This is particularly important at night when visibility is reduced.
  There may be tools and equipment on the ground that you might fall over. There may be open that you might fall into deep, trenches sometimes contain poisonous gases that are heavier than air.

RADIATION HAZARD

- Many SGC plants use radiation test equipment to X-ray pipelines to check the thickness of the metal.
- The warning sign shown in figure 25 will be displayed. Whenever the test equipment is used in a plant, the testing area will be taped off and the radiation warning sign displayed.

Figure 25: Radiation Warning Sign

- Long periods of exposure to radiation can be dangerous to you. Radiation can cause cancer. That is why radiation sources are stored away from plant personnel.
- In you normal duties in a plant, you will not be in any danger of receiving hazardous doses of radiation, but learn to recognise the warning sign.

PLANT VEHICLE HAZARDS

- For safety reasons, vehicles are allowed to be driven only on designated roads inside Petroleum Companies facilities. Entry into restricted areas (process plant areas) is controlled by Petroleum Companies Work Permit procedures.
- All internal combustion engines produce sparks or hot spots, which could ignite gas that might be in the air. Vehicles allowed into Petroleum Companies facilities are inspected regularly. If a car truck runs into a pipeline or vessel, the vehicle can break the pipeline or vessel open. The result could be a major oil spill, or the releases of toxic or explosive gas.
- All vehicles driven inside a plant must be operated at very low speeds. A speeding vehicle is always a hazard. Inside a hydrocarbon plant, it can be deadly.
- One of the most dangerous vehicle hazards in a plant is when a vehicle is backing up without a guide. Even a slow moving vehicle that backs into a pipeline, or a vessel, or their support columns, can cause major damage and a major hazard.

Speed Limits

Posted speed limits must be observed at all times while driving SGC vehicles or on SGC property.

Normal *maximum* vehicle speeds are:

- Base: 20 km/hr
- Well access roads: 50 km/hr
- Main road: 80 km/hr

As weather or road or construction conditions dictate, maximum allowable speeds can be lower than those posted above.

Passengers

Passengers shall only be carried in vehicles that are designed for the purpose.

Sitting on fenders or tailgates of moving vehicles is prohibited. No person is allowed to ride inside the cargo box of any truck but must be seated in the passenger compartment.

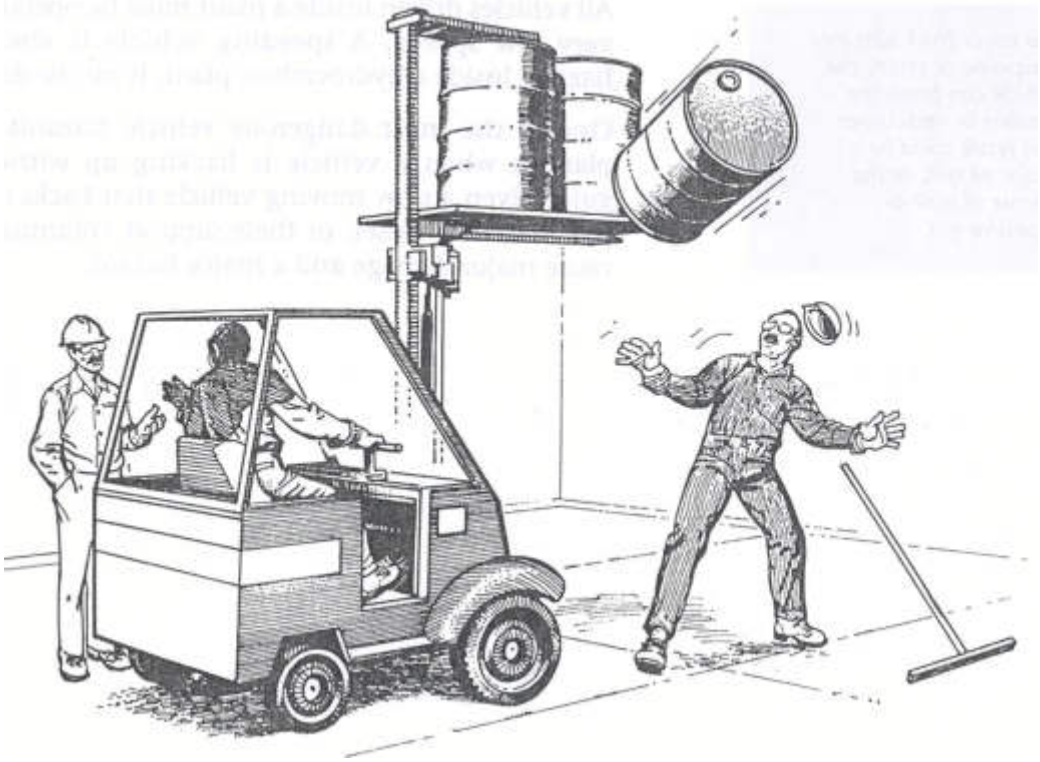
Seat Belts

Seat belts **must** be worn at all times by all vehicle occupants in SGC or Contractor vehicles. Those not wearing seat belts could endanger others, should they become involved in an accident.

OVERHEAD WORK HAZARDS

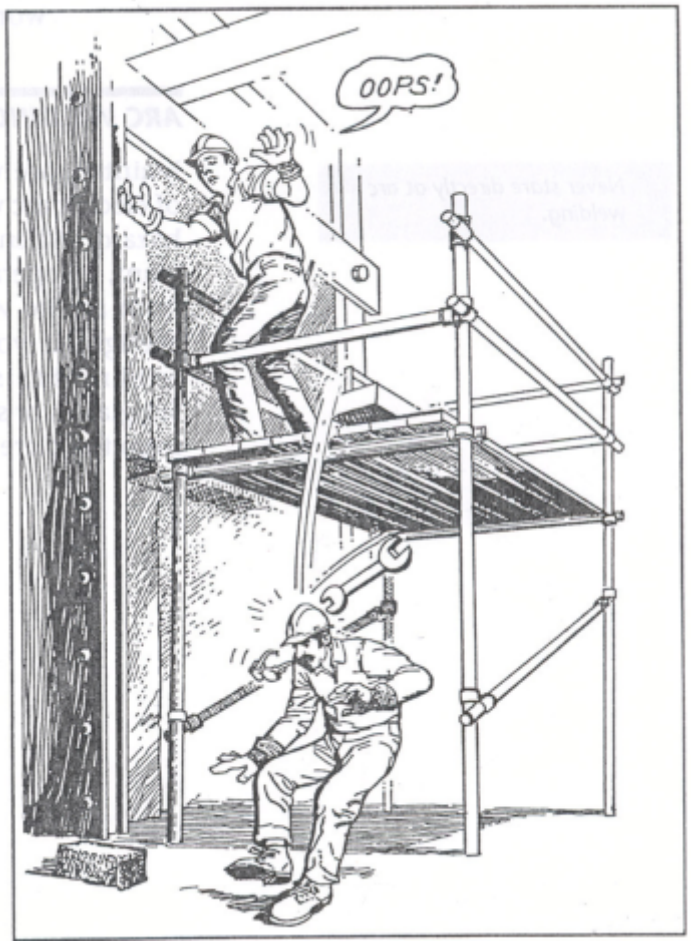
- Any load hanging from a crane or being moved by a fork truck is a potential hazard. Common sense tells you not to walk or stand under a load.
- Be alert when cranes are working in your area. A heavy load does not have to be very high above you to be a hazard.

Figure 26: Carelessness Causes Accidents.



- Large, heavy overhead loads are easy to see and avoid as long as you keep alert.
- Small objects that fall from overhead can also be a hazard. Maintenance work is often performed on elevated platforms and scaffolds. A workman may drop one of his tools. He may drop a replacement part he is installing. A nut or bolt may fall. Any of these objects dropped from a height becomes a dangerous missile.

Figure 27: Watch Out Below



- There are several ways to protect yourself against accidents from falling objects:
  - **Always wear your hard hat. It is designed to protect your head.**
  - **Know what is going on in the plant. If overhead maintenance work is being performed in your area, you must know about it.**
  - **If you pass under an area where men are working, be alert for falling objects.**

ARC WELDING

- Maintenance work in a plant often requires the use of electric arc welding machines.
- Arc welding is a fire hazard. When arc welding is being performed in a plant, an operator is assigned as fire watch.
- The glare from an arc welder can permanently damage your eyesight if you stare at it. The worker doing the welding has special eye protection. You do not.
- Sunglasses or safety glasses will not protect your eyes from the glare of an arc-welding machine.

- **Never stare directly at arc welding**

Figure 28: Never stare directly arc welding

LIFTING HAZARDS

- In your work as a plant operator, you may have to lift heavy objects. Bruises, cuts, punctures, or even crushed fingers and hands can happen can if you are not careful when you move something from one place to another.

- Most lifting injures are to the lower back. There is a proper way to lift objects. Figure 28 shows you how to do it without hurting your back. Bend your knees, but keep your back straight.

Figure 29: Load Lifting

- **Make sure to use your legs to lift heavy objects, not your back.**

PRACTICING SAFETY

- Always practice safety when working in a plant. Stay alert and use your common sense. The hazards are there, but they only cause accidents if you let them. Recognise hazards. If you are not sure about something, discuss it with your Supervisor. Don't take chances.

SAFETY SIGNS STANDARDS

The information and instructions in the Table below apply to all safety signs.

Colour Coding

COLOUR	MEANING OR PURPOSE	INSTRUCTIONS
RED	Prohibition	Avoid dangerous actions
	Danger - Alarm	Stop, shut-down, turn off power, evacuate.
	Fire-Fighting Equipment	Identification and location
YELLOW	Warning	Be careful, take precautions, examine
BLUE	Obligation	Prescription of behaviour or action. Obligation to wear personal protective equipment
GREEN	Emergency escape routes First-Aid	Doors, exits, routes Equipment, identification, location
	No danger	Back to normal

A safety sign refers to a specific object, activity or situation and provides information about a hazard or danger to health and safety at work, a prohibition, a mandatory action, an escape route or the location of fire-fighting and first aid equipment.

The placing of too many signs too close together must be avoided. Appropriate safety signs must be clean, maintained or replaced (if necessary) at appropriate intervals. A safety sign must be removed when the situation to which it refers no longer exists.

Prohibition Signs

Intrinsic features:
Circular shape, black pictogram on white background, red edging, red diagonal line.

No smoking

Smoking and open flames forbidden

Do not extinguish with water

No access for industrial vehicles

Do not touch

No pacemakers

Warning Signs

Intrinsic features:

Triangular shape, black pictogram on yellow background, black edging.

Flammable material or high temperature

Harmful or irritant material

Toxic material

Corrosive material

Radiation or radioactive material

Overhead load

Laser

Non ionizing radiation

Strong magnetic field

Obstacles

Drop

Slippery surface

Industrial vehicle

Danger electricity

Mandatory Signs

Intrinsic features:
Round shape, white pictogram on blue background.

Eye protection must be worn

Safety helmet must be worn

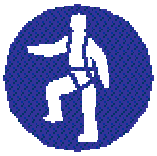
Ear protection must be worn

Respiratory equipment must be worn

Safety boots must be worn

Safety gloves must be worn

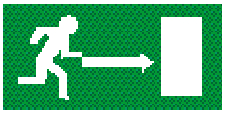
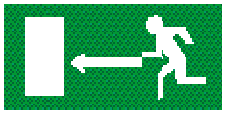
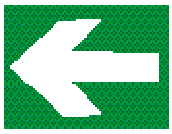
Face protection must be worn



Safety harness must be worn

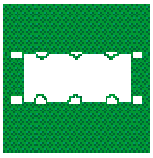
Emergency Escapes / First-Aid Signs

Intrinsic features:
Rectangular or square shape, white pictogram on green background.

This way (supplementary sign)

First aid post

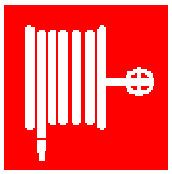

Stretcher

Safety shower

Eyewash

Fire-Fighting Signs

Intrinsic features:
Rectangular or square shape, white pictogram on red background.



Fire Hose

Fire Extinguisher

Radiation

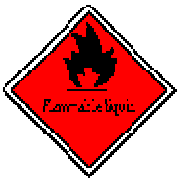
Warning panel for high radiation sectors

Warning panels for areas where radioactive contamination is possible

This sign is displayed on vehicles carrying radioactive or irradiated material.

Flammable Gases and Chemical Materials

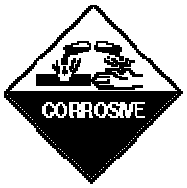
Flammable Gas



Flammable Liquid

Flammable Material

Poisonous Substance

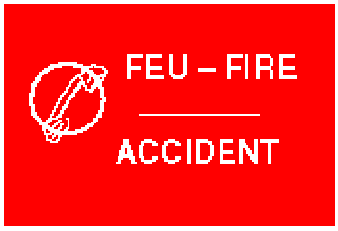


Corrosive Substance

General Signs

Prohibitory signs for strong magnetic fields

No cleaning



To identify emergency (red) telephones

IDENTIFY UNSAFE ACTS

- Accidents do not happen by themselves. People cause them.
- There are many reasons why people cause accidents; carelessness, haste, fatigue, ignorance, laziness and over-confidence are some of the reasons.
- Petroleum Companies has safety rules for all plants. These rules have been used for many years. Safety rules and procedures are not only for plant operators. They apply to everyone who works in or enters Petroleum Companies plants. Safety is everyone's business.
- In this part of the module, you will see some examples of unsafe acts that can lead to accidents. They are:
  - Unsafe contact with leaking substance
  - Working outside a safety railing
  - Failure to replace a machinery guard
  - Ignoring an oil spill
  - Improper vehicle operation
  - Failure to wear safety equipment
  - Improper lifting
  - Smoking in restricted area
  - Doing unauthorised work
  - Inviting an accident

UNSAFE CONTACT WITH LEAKING SUBSTANCE

- The operator in Figure 30 could end up with a badly burned or permanently disabled hand. The substance leaking from the pipe could be a hot or highly corrosive liquid.
- **Avoid skin contact with leaking liquids.**

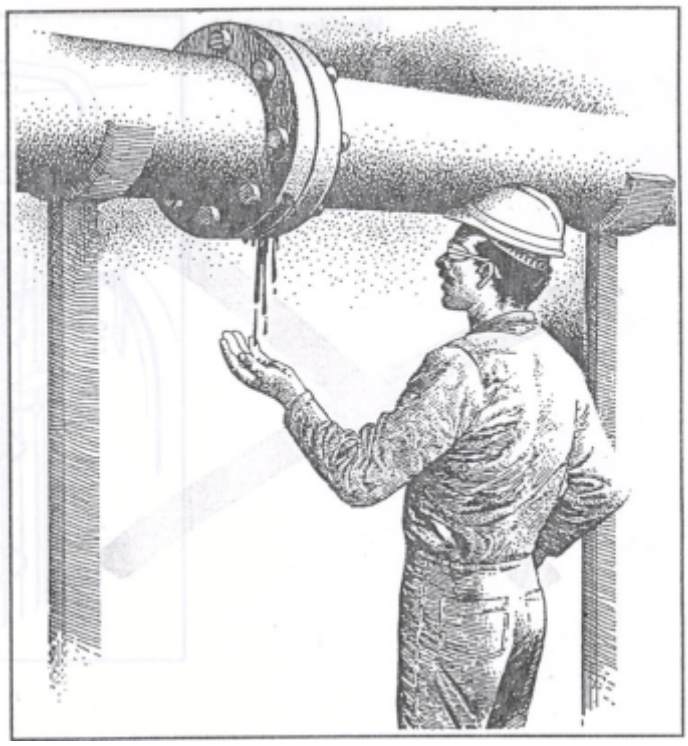


Figure 30 : Unsafe Contact with Leaking Substance

WORKING OUTSIDE A SAFETY RAILING

- The operator in Figure 31 is working on an elevated platform. The valve he is operating is out of reach from inside the safety railing. He has climbed over the railing. If this hand slips, there is nothing to stop him from falling.
- **Stay inside the safety railing. If you have to go outside the safety railing, you must wear a safety belt.**

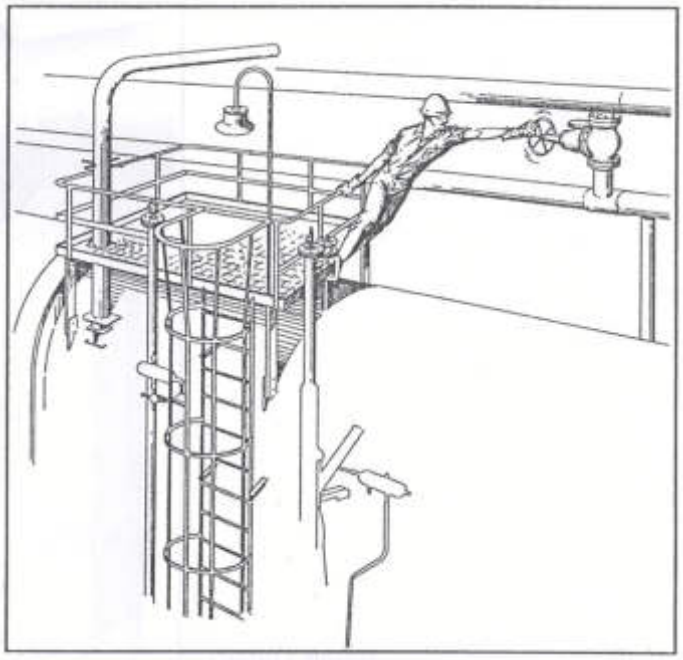


Figure 31: Working Outside a Safety Railing

FAILURE TO REPLACE A MACHINERY GUARD

- In Figure 32, someone has removed the guard to do some maintenance work on a machine and forgotten to replace it. Maintenance personnel should never leave a piece of rotating equipment before all covers and guards have been reinstalled.
- Make it a habit to check the condition of machinery in your plant. Check for covers and guards that may have been left off machines.

- **Report missing machinery guards to your supervising operator immediately.**

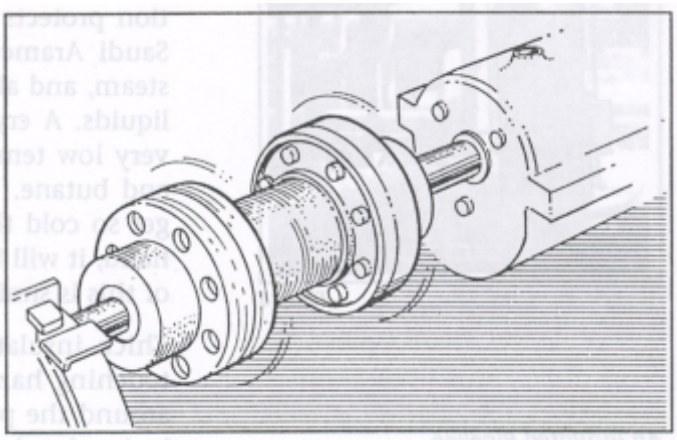


Figure 32: Failure to Replace a Machinery Guard

IGNORING AN OIL SPILL

- A possible result of ignoring an oil spill is illustrated in Figure 33. The real mistake here was in not cleaning up the spill when it happened.
- If walking through an oil spill is unavoidable, walk slowly and carefully.
- **Clean up oil spills as soon as they happen. Tape off the area until the spill can be cleaned up.**

Figure 33: Ignoring an Oil Spill

IMPROPER VEHICILE OPERATION

- The illustration in Figure 34 shows an operator backing a vehicle without a guide; He could back into a pipeline, a vessel, a fire hydrant or monitor, or a support column, causing manor damage and major hazard.
- **Always have someone guide you when backing up a vehicle in the vicinity of plant equipment.**

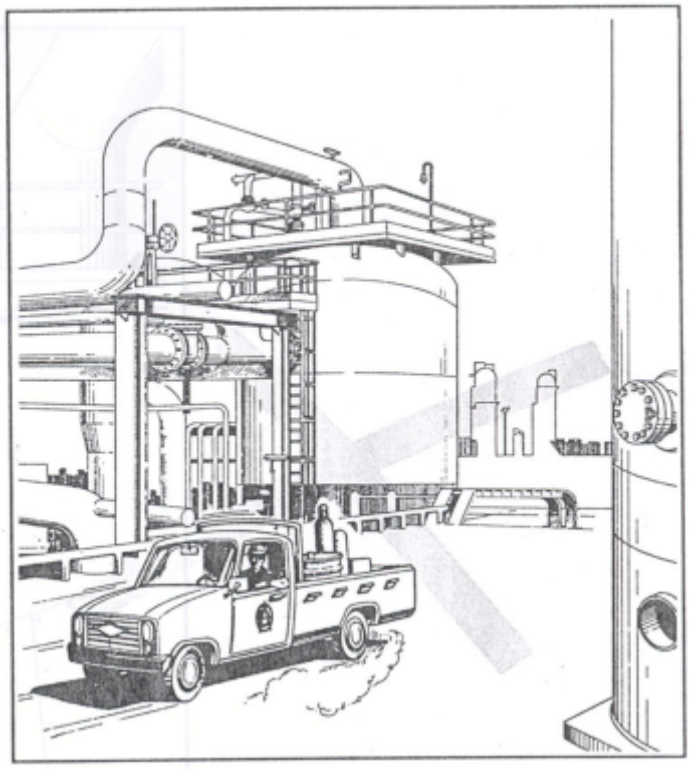


Figure 34: Backing Up Without a Guide

FAILURE TO WEAR SAFETY EQUIPMENT

- Beware of objects falling from above. Even small objects can strike with great force if they fall from a great height. Pointed objects can be deadly. The operator in Figure 35 is exposing himself to a hazard by removing his hard hat.
- **Always wear your safety hat when outside in the plant.**

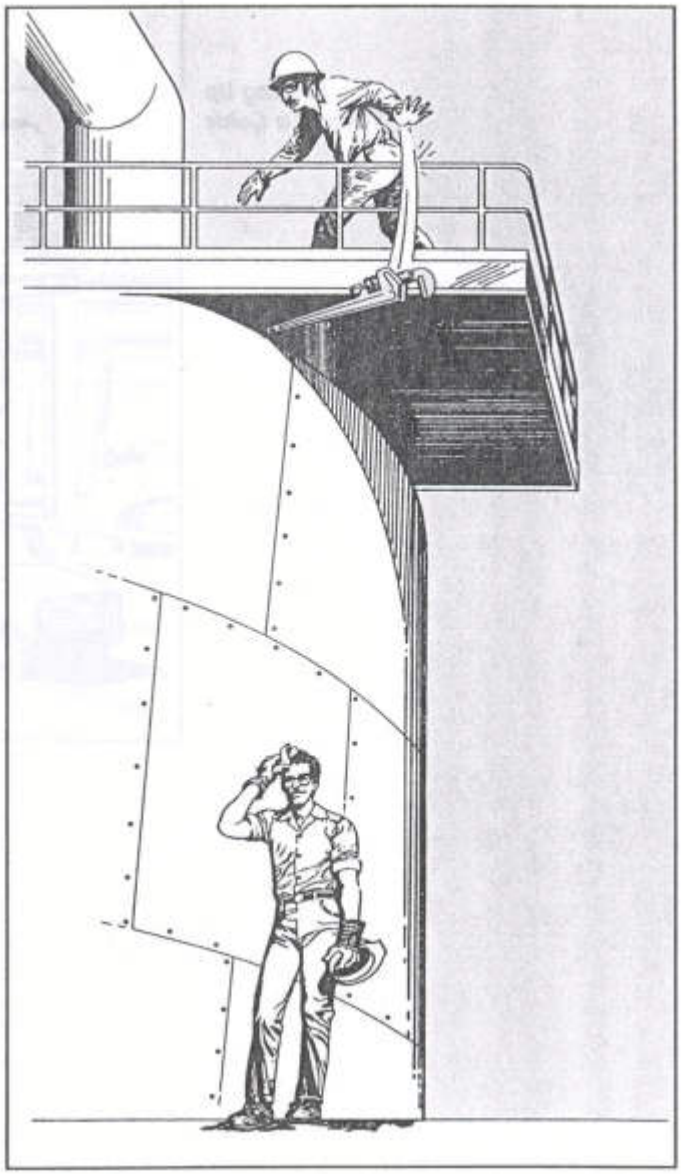
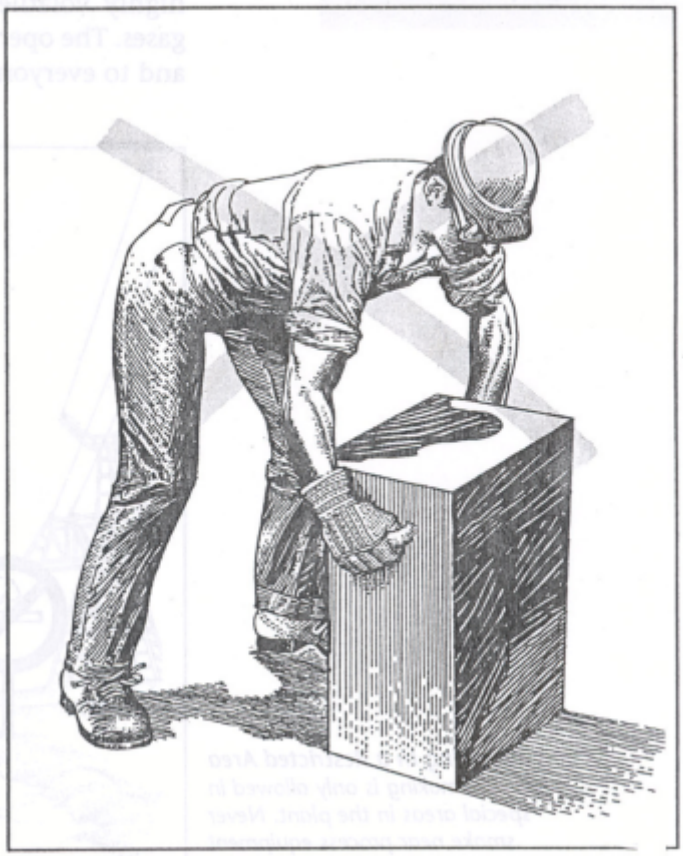


Figure 35: Failure to Wear Required Safety Equipment

IMPROPER LIFTING

- This operator in Figure 36 does not know the proper way to lift a heavy object his lack of knowledge may result in a back injury.
- **Always follow correct lifting procedures.**

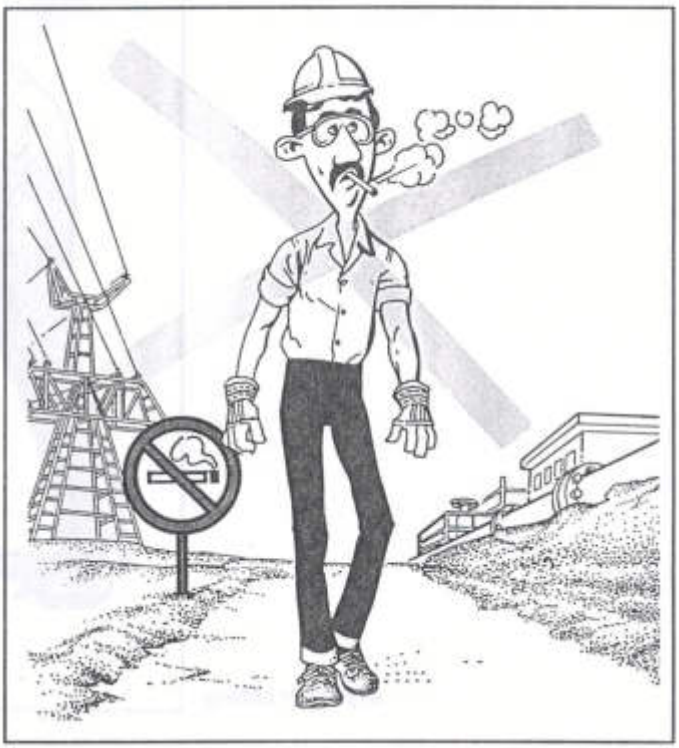
Figure 36: Improper Lifting



SMOKING IN RESTRICTED AREAS

- You are not allowed to smoke cigarettes, cigars, or pipes in Petroleum Companies plants, except at safe places identified as smoking areas.
- The reason smoking is prohibited is obvious. Petroleum Companies plants process highly volatile substances that produce flammable gases.
- The operator in Figure 37 is a danger to himself and everyone else in the plant.
- **Smoking is designated in certain safe plant smoking areas. Be sure to extinguish your cigarette and bury it in the sand stand provided.**

Figure 37: Smoking in a Restricted Area



Smoking is only allowed in special areas in the plant.
Never smoke near process equipment or in the control room.

DOING UNAUTHORIZED WORK

- Look at the plant operator in Figure 38. He is doing work he should not do. He has not been trained to work on process instruments.
- Always call maintenance to do repair work. The maintenance department has trained specialists to keep plant equipment running correctly.
- These are some of the things that could happen because of the operator's actions:
  - The man could receive an electric shock.
  - A plant process could be upset.
  - The instrument could be damaged.
  - The man's action could cause the plant to go into an emergency shutdown.

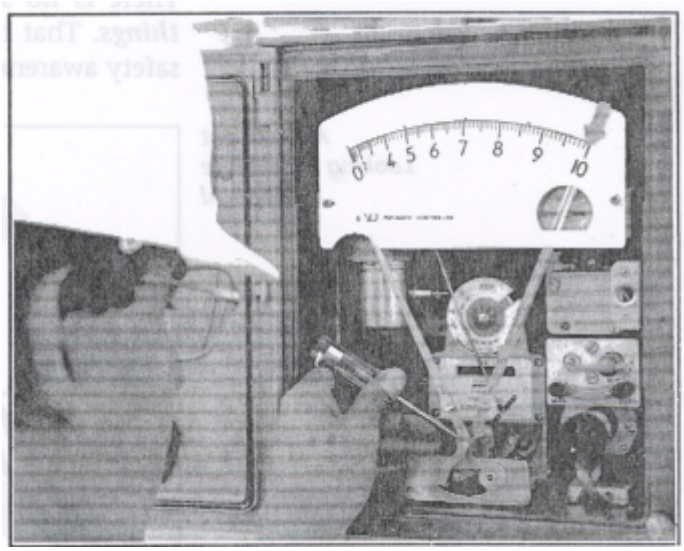


Figure 38: Leave Maintenance Work to Maintenance Personnel

INVITING AN ACCIDENT

- Look at the man in Figure 39. He is in a hurry. The man is not concerned for his safety. This may be the fastest way to cross these pipelines. It is not the safe way to do it.
- A pipeline is a curved surface. The bottom of the shoes may be wet, or worse, may have oil or grease on them. The pipe may have oil on it. Even water on the pipe can be slippery. One false step and the result could be a broken leg.
- There is no safety rule that says **don't do foolish things.** That has to come from within you. That is safety awareness.

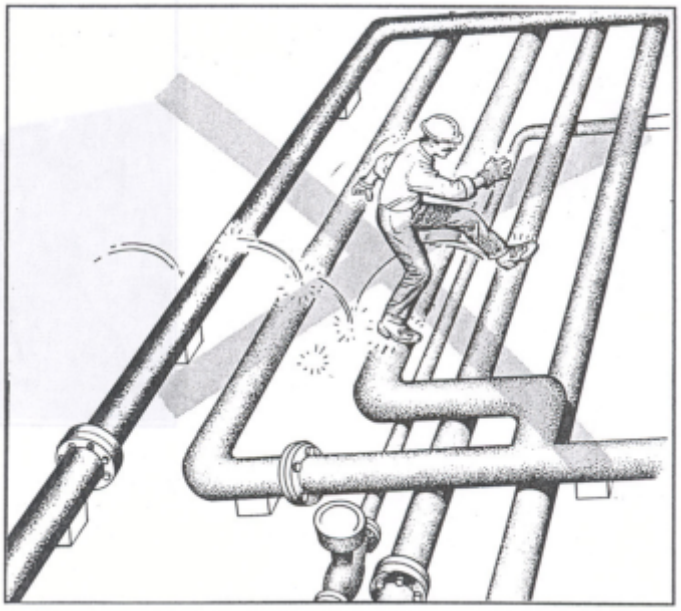


Figure 39: An Accident looking For a Place to Happen!

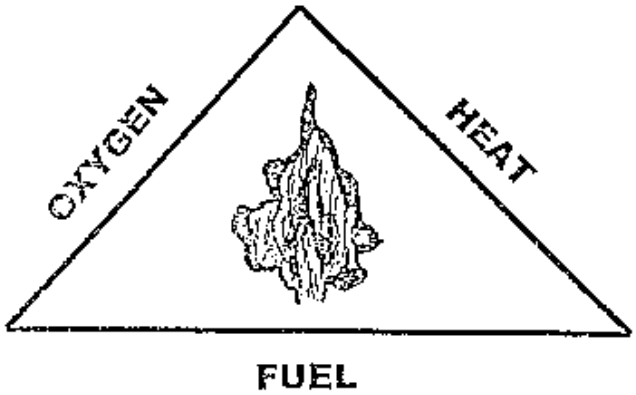
FIRE

Definition of fire:

Fire is a chemical reaction, which produces, **Heat, Light and Noise**

Fire Triangle:

Three things are needed for fire to take place:



- Fuel, this can be in the form of a gas, vapour or dust
- Heat, this can be an arc, spark, naked flame or even a hot surface.
- Oxygen plentiful supply since there is approximately 21% by volume air.

CLASSES OF FIRE

Fire is classified as follows:

Class "A"	Solids (Wood, Paper, Textiles, Furniture and People)
Class "B"	Flammable Liquids and Gases (Paint, Kerosene, Oil, Grease, Fats, Waxes and Gasoline)
Class "C"	Electrical (Motors, generators, switch panels and computers)
Class "D"	combustible Metals (Sodium, Magnesium, Potassium and zirconium)

Types of Extinguishers

Type	Code Color	When to use
Water	RED	Class "A" Fire. Don't use on life electrical equipment.
Dry Powder		Class " D" Fire.
Foam		Class " B" Fire.
Carbon Dioxide		Class " B & C" Fires.
Dry Chemical		Class " B & C" Fires.

Ignition Temperature:

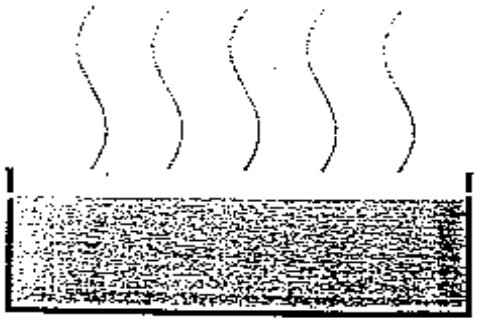
Ignition Temperature is the minimum temperature at which a material will spontaneously ignite.

Flash Point:

Material	Ignition Temperature °C
Propane	470
Ethylene	425
Hydrogen	560
Acetylene	305
Diethyl Ether	170
Paraffin	210

Flash point is the lowest temperature at which sufficient vapour is given of f a liquid, to form a flammable mixture with air in the space above the liquid. This mixture can be ignited by an arc, spark or naked flame. The flash temperature of liquid is usually much lower than the temperature at which the liquid boils.

Material	Flashpoint °C
Propane	-104
Ethylene	-120
Hydrogen	-256
Acetylene	- 82
Diethyl Ether	- 45
Paraffin	38

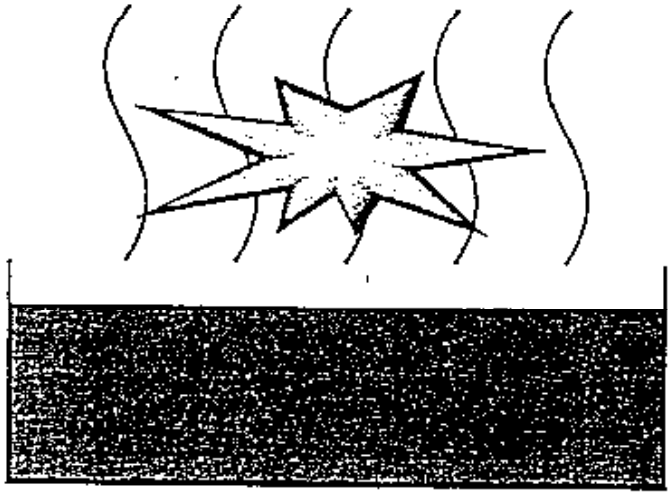


Vapour Quantity Dependent on Temperature

Flashpoint

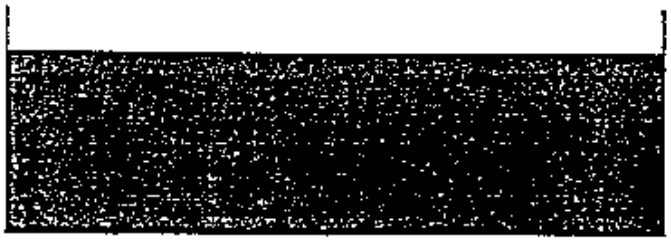
Kerosene (Flashpoint 38°C)

at 38°C
"IGNITION"



at 37°C
INSUFFICIENT VAPOUR
GIVEN OFF

at 0°C
NEGLIGIBLE VAPOUR
RELEASED

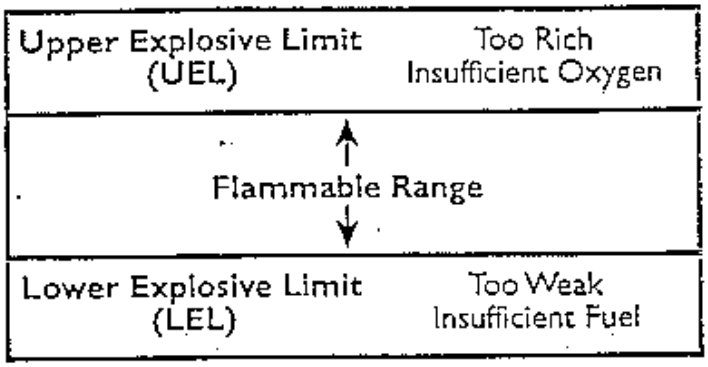


Explosive Limits

The gas/air mixture must fall within certain limits, which are the 'lower explosive limit' (LEL), and the 'upper explosive limit' (UEL), before combustion can take place. An every day example of this action is the carburettor of a petrol engine, which must be tuned to a particular point between these limits in order that the engine may function efficiently.

Lower Explosive Limit (LEL): -

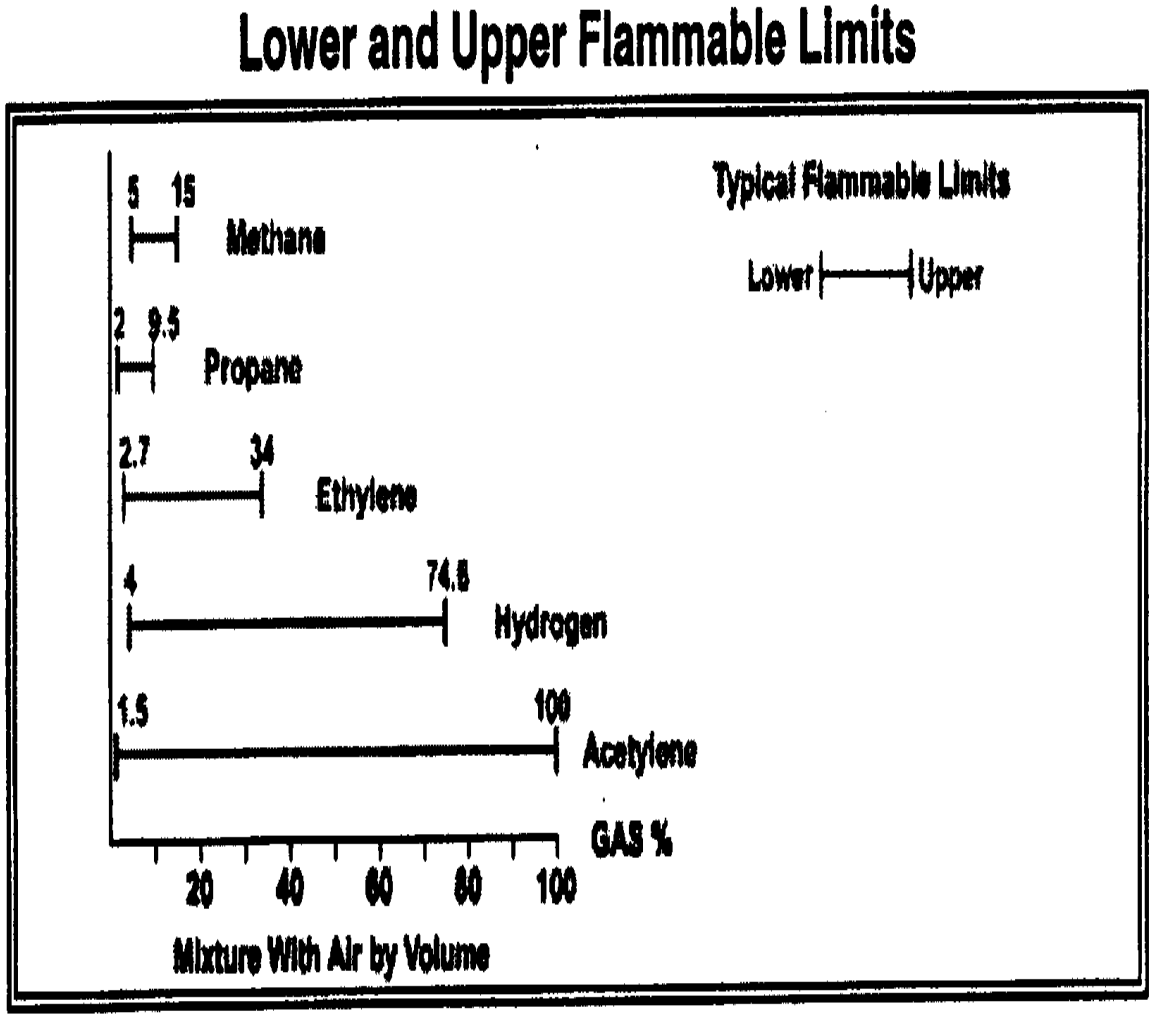
The concentration of flammable in air (%) below which the mixture will not support



combustion.

Upper Explosive Limit (UEL): -

The concentration of flammable in air (%) above which the mixture will not support combustion.



LOWER AND UPPER EXPLOSIVE LIMITS

Classification of Hazardous areas

Hazardous areas: -

Hazardous areas are those areas where an explosive gas- air mixture is present or most likely to occur.

Classification of Hazardous areas:

These areas are classified in European standards as follows: -

Zone 0 : An area in which an explosive gas atmosphere is present continuously or for long periods. (Over 1000 hours per year).
Example: tank vents, sump vents, drains etc.

Zone 1 : An area in which an explosive gas atmosphere is likely to occur in normal operation (10-1000 hours per year).
Example: pipeline flanges and inlet manifold areas.

Zone 2: An area in which an explosive gas atmosphere is likely to occur in abnormal operation (0-10 hours per year).
Example: welded pipelines where there are no flange joints.

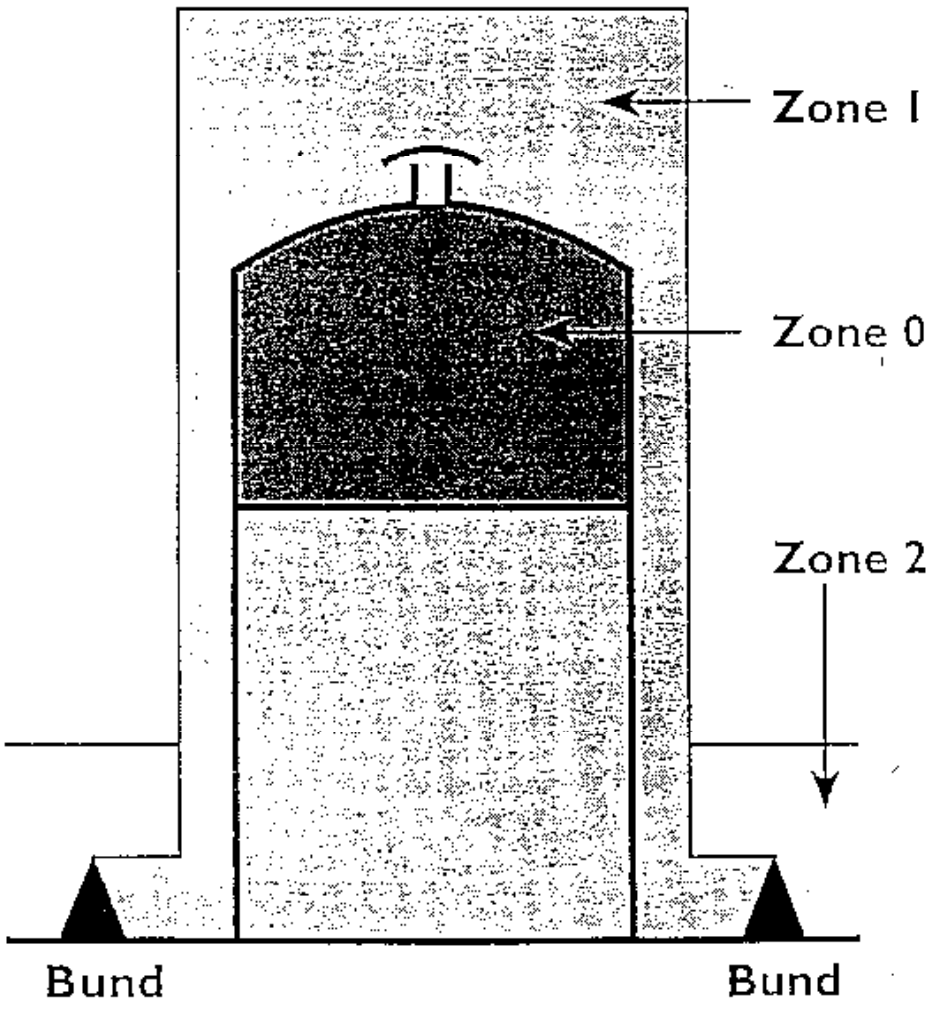
Non Hazardous Area : An area in which an explosive gas atmosphere is not expected to be present in quantities such as to require special precautions.

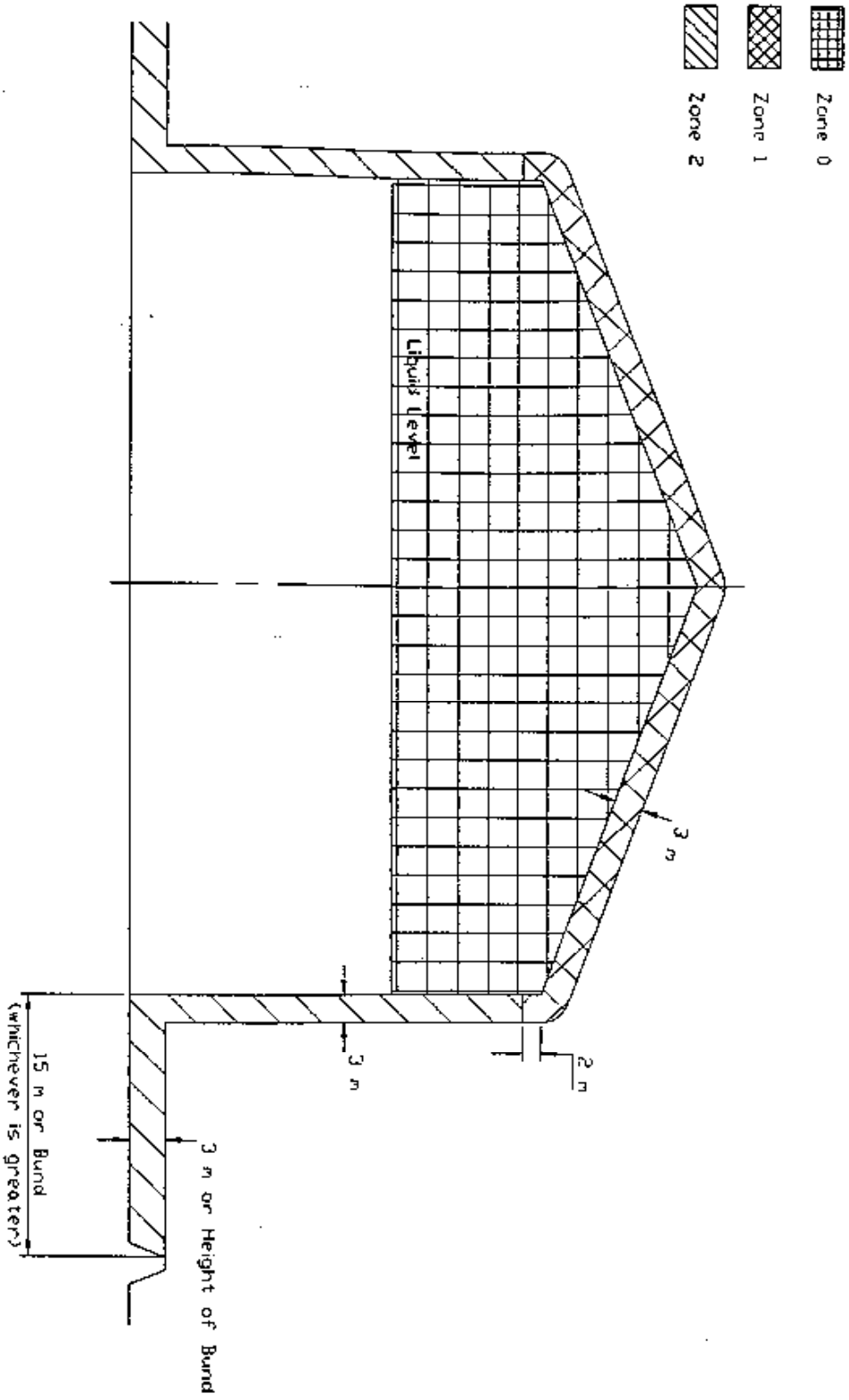
Purpose of area classification:

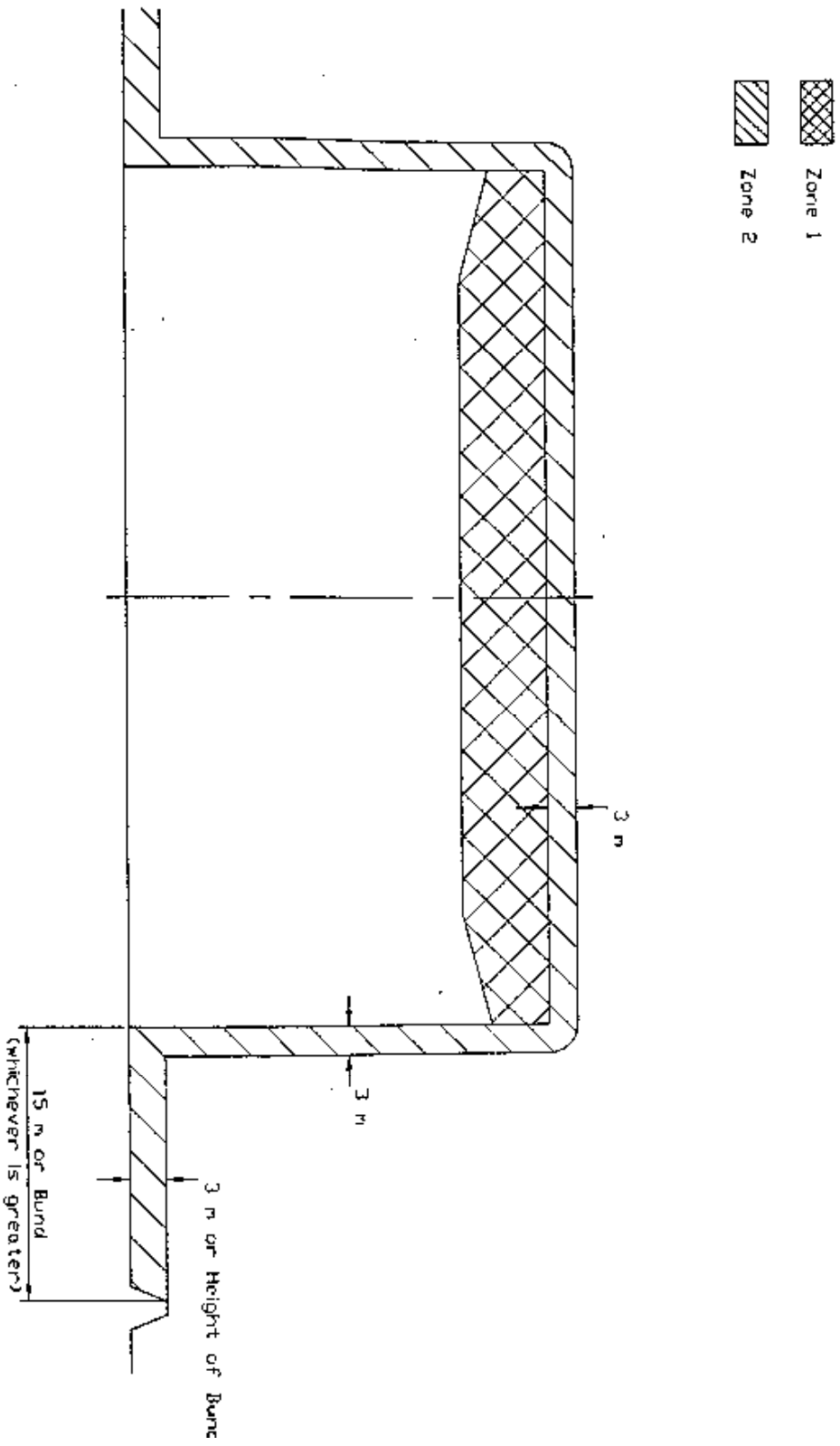
- Selection of electrical equipment.
- Consideration of all sources of ignition.
- Permits to works.
- Access of portable equipment.

Area Classification

FIXED ROOF STORAGE TANK




Zone 0
Zone 1
Zone 2
Liquid Level
3 m
3 m
2 m
3 m or Height of Bund
15 m or Bund
(whichever is greater)


Zone 1
Zone 2
3 m
3 m
3 m or Height of Bund
15 m or Bund
(whichever is greater)

Zone 0
Zone 2
3 m radius

WORK PERMITS

Work Permit:

A work permit shall be raised to cover all activities, other than routine operations**.** The Permit to Work system is SGC' principal safety protection for both the work force and facilities, while carrying out any form of potentially hazardous work.

The Permit to Work system is used to:

- Initiate work.
- Communicate work tasks and precautions.
- Prepare equipment for work.
- Prepare equipment for return to operational mode.

The Permit to Work system covers:

- **Hot Work** – that works that could provide sources of ignition in areas where explosives / flammable gases or liquids exist.
- **Cold Work** – which is any work providing a potential hazard to personnel and equipment, other than Hot Work.
- **Confined Space Entry** – where workers may perform tasks in an enclosed space, which may be hazardous due to a toxic or oxygen deficient/rich atmosphere.
- **Excavation Work** – where the excavating will be performed at existing facilities, or whenever an underground structure may be encountered.

Glossary

Article: A piece; one piece
Assembly Area: A designated place to go and meet in emergency.
Common Sense: Good judgment.
Corrosive: Having the power to eat or wear away, usually by chemical action.
Crane: A machine for lifting, moving, and lowering heavy weights by means of a projecting boom, or arm.
Demonstrate (to): To do; to show how to do
Diagram: A drawing
Evacuate (to): To get out; to remove from
Facility: Something that's built to serve a particular purpose
Flange: A rib or rim provided for strength or for attaching to another object
Industrial Area: Plants; workshops
Nonflammable: Will not burn
Operate (to): To switch on; to run
Outlet: A socket where an electric cord is plugged in
Particles: Small pieces
Permanent: Fixed, forever
Potable: Clean enough to drink; as potable water.
Power: A force that does work, as human, animal, or electrical
Prescribed: Ordered
Procedure: An orderly way to do something
Rigid: Stiff, Hard
Rotating: Turning, spinning.
Rung: One of the crosspieces of a ladder.
Scaffold: A platform at a height above the ground
Site: Location
Supply (to): To give to; to provide. A generator supplies electric power.
System: A group of parts that operate together to perform a job
Tread: The pattern on a tire, or the sole of a boot of shoe, that gives a good grip
Vital: Important
Vibration: Rapid, continuous motions back and forth or up and down
Hazards: Dangers
Flammable: Capable of burning.